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1050 K Stirling Space Engine Design

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1050 K STIRLING SPACE ENGINE DESIGN

SUMMARY

As part of the NASA CSTI High Capacity Power Program on Conversion Systems for Nuclear Applications, Sunpower, Inc., completed under contract for NASA Lewis a reference design of a single-cylinder free-piston Stirling engine that is optimized for the lifetimes (seven years) and temperatures appropriate for space applications. The NASA effort is part of the overall SP-100 program which is a combined DOD/DOE/NASA program to develop nuclear power for space. Stirling engines have been identified as a growth option for the SP-100 power system offering increased power output and lower system mass and radiator area.

Nickel-based superalloys are the primary materials used in the Stirling-cycle portion of the engine. The hot-end temperature of the engine is 1050 K (helium-side heater wall temperature); the engine temperature ratio is 2.0. The engine design features simplified heat exchangers with heat input by sodium heat pipes, hydrodynamic gas bearings, a permanent magnet linear alternator, and a dynamic balance system. Heat is rejected to a pumped NaK loop. The design shows an efficiency (including alternator) of 29 percent and a specific mass of 5.7 kg/kW(e). This design also represents a significant step toward the 1300 K refractory Stirling engine which is a growth option for SP-100.

INTRODUCTION

This superalloy Stirling Space Engine (SSE) reference design represents the results of an extensive effort by Sunpower, Inc., under contract for NASA Lewis, to investigate the applicability of free-piston Stirling engine (FPSE) technology for a space dynamic power conversion system. The current design is an offshoot of the initial Stirling engine efforts done in support of the SP-100 program. SP-100 is a combined DOD/DOE/NASA program to develop nuclear power for space.

The free-piston Stirling engine has been identified as having several very attractive features for space power applications. It has a high efficiency and can achieve that efficiency at low engine temperature ratios. Low temperature ratios and high efficiencies allow power system operation at high cold-end temperatures for lower radiator mass and area or possible use of a lower-temperature reactor. FPSE values of efficiency and specific mass allow minimizing overall system mass compared to other power conversion systems. Having only two to three moving parts per cylinder, utilizing noncontacting gas bearings on these moving parts, and using a hermetically sealed system give the potential for long life and high reliability. The FPSE can be dynamically balanced in either single-cylinder or opposed-piston configurations.

Stirling engine technology development under the SP-100 program began in the 1983 to 1984 time frame with contracted efforts at Sunpower, Inc., and Mechanical Technology, Inc. (MTI). MTI designed, fabricated, and tested the opposed-piston 25 kW Space Power Demonstrator Engine

(SPDE) to show scaling to larger power levels and engine operation at a temperature ratio of 2.0. The SPDE had a hot-end temperature of 650 K. The SPDE is described in reference 1.

Sunpower, Inc., first completed the conceptual design of a single-cylinder 25 kW engine optimized for the lifetimes and temperatures appropriate for a space power system. This engine had a temperature ratio of 2.0 but with a hot-end temperature of about 1100 K. The design used a pumped sodium loop on the hot end and a pumped NaK loop on the cold end; these heat transport systems and the engine temperatures were chosen to correspond with the SP-100 Stirling engine power system designs that were being studied at that time. The primary hot-end material was a refractory metal which was required at these temperatures. In support of this design, Sunpower demonstrated in hardware tests the feasibility of two key component technologies - hydrodynamic gas bearings and a dynamic balance system. Also completed were parametric analyses to define the Stirling engine specific mass and efficiency for various temperature ratios. This phase I work done by Sunpower is reported in reference 2.

Following this initial phase, Stirling engine technology efforts have continued under the NASA CSTI High Capacity Power Program on Conversion Systems for Nuclear Applications, a part of the SP-100 program. Free-piston Stirling engines have now been identified as a growth option for SP-100 offering increased power output and lower system mass and radiator area. The Sunpower design reported in this paper was initiated at this time. The objective was to modify the phase I design to use superalloy materials near their maximum operating temperature; superalloy materials would allow easier fabrication and testing compared to the refractory materials. The other main objective for the design was to investigate the possibilities of using heat pipes, especially for the hot-end heat transport system.

The superalloy SSE design focused on a hot-end temperature of 1050 K. The temperature ratio is 2.0, a value that optimizes the overall nuclear power system for minimum mass. The engine design features simplified heat exchangers with heat input by sodium heat pipes, hydrodynamic gas bearings, a permanent magnet linear alternator, and a dynamic balance system.. The design demonstrates specific mass, efficiency, and potential reliability that make it a viable candidate for space power systems at its design temperatures. This reference design also represents a significant step toward the design of a 1300 K refractory Stirling engine which is a growth option for SP-100.

Other work is continuing at Sunpower to demonstrate the feasibility of the simplified heat exchanger modules that are being utilized in the engine design. Flow tests are being conducted in an oscillating flow rig capable of operation up to 120 hertz oscillating frequency to determine pressure drop through the heat exchangers. This oscillating flow rig is described in reference 3. Also, a heater head using three heat exchanger modules heated by sodium heat pipes is being fabricated for a 1 kW research engine on test at NASA Lewis. This heater head design and fabrication is discussed in reference 4.

STIRLING SPACE ENGINE POWER MODULE DESIGN

General

The primary power module reference design criteria are listed in Table 1.

Table 1: Power Module Reference Design Criteria

- Single Cylinder / Linear Alternator
  Helium
- 25 kW(e) Alternator AC Output
- Design Life — 60,000 Hours
- Superalloy Hot End
- T_H — 1050 K T_C — 525 K
- Heat Transport Hot End — Heat Pipe
- Cold End — Pumped Loop
- Demonstrate Low Specific Mass

The key components making up the FPSE power module are depicted in Figure 1. In its current configuration the power module represents a conventional single cylinder FPSE employing a permanent magnet linear alternator for power production. Unique to this configuration is a dynamic balance unit which is employed to counteract the unbalanced forces on the engine casing due to the motion of the power piston and displacer. Maintaining low engine system specific mass in conjunction with high thermal performance at the low operating temperature ratios resulted in engine operating conditions, shown in Table 2, well in excess of conventional FPSEs. The combination of high operating pressures and temperatures plays a major role in determining the hot-end component configuration and as such represents one of the critical technology areas in the current system.

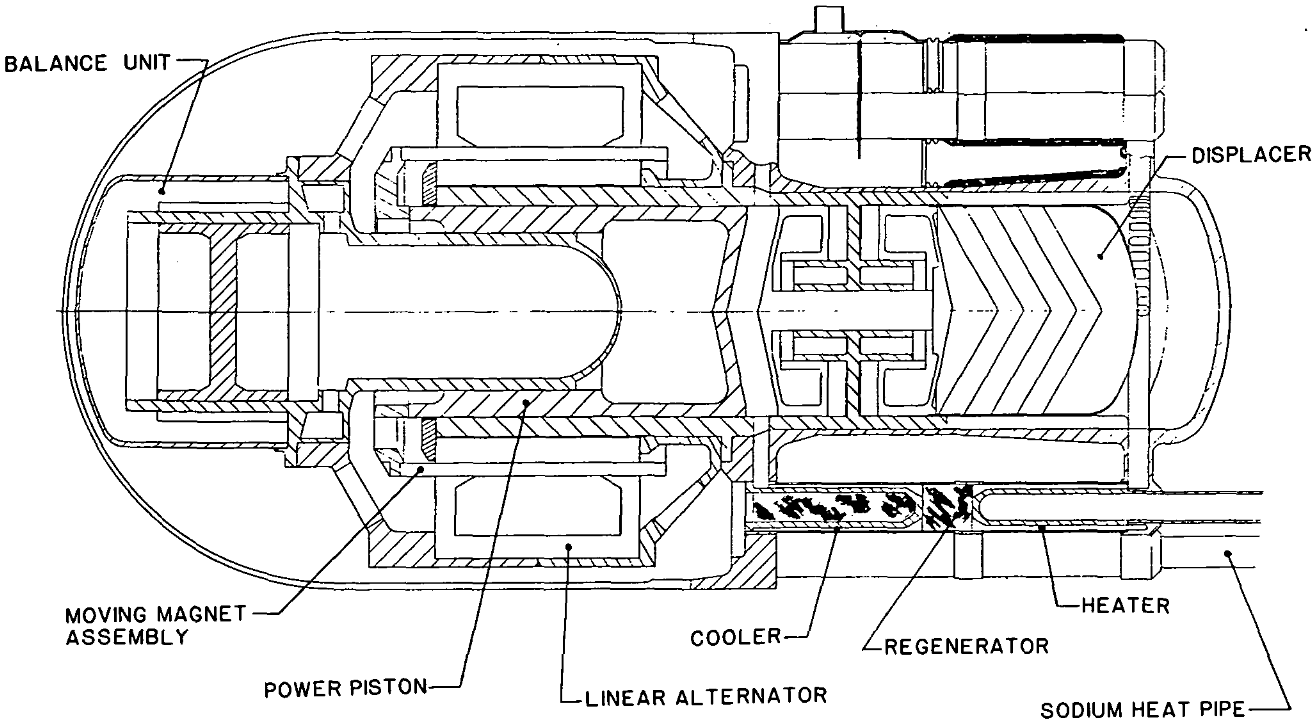

Figure 1: Reference Superalloy Power Module

Table 2: Power Module Nominal Operating Point

Engine Mean Pressure	151 bar
Operating Frequency	90 Hz
Piston Amplitude	15 mm
Displacer Amplitude	12 mm
Pressure Amplitude	16.3 bar
Pressure Phase Angle	13.7 degrees
Piston-Displacer Phase Angle	44.9 degrees
Engine Hot-End Temperature	1050 K
Engine Cold-End Temperature	525 K
Regenerator Temperature Ratio	1.88
Electric Power Output	27 kW(e)
Power Module Efficiency	0.289
System Specific Mass	5.7 kg / kW(e)

Table 3: Basic Stirling Cycle Characteristics

Power

Cycle PV Power	32 kW	
•Piston Seal Leakage Loss		126 W
•Displacer Gas Spring Hysteresis		358 W
•Displacer Gas Spring Leakage Loss		445 W
•Displacer Appendix Gap Loss		450 W
•Balance Unit Loss		690 W
Basic Piston PV Power	30 kW	
•Piston Gas Spring Leakage Loss		148 W
•Piston Gas Spring Hysteresis		174 W
Power to Alternator	29.7 kW	
Alternator Electric Power Output	27 kW	

Thermal

Basic Cycle Thermal Input	91 kW	
•Pressure Wall Conduction		240 W
•Cylinder Wall Conduction		45 W
•Displacer Shell Conduction		82 W
•Regenerator Wall Conduction		1755 W
•Regenerator Gas Conduction		290 W
•Displacer Shuttle Heat Transfer		75 W
Adjusted Cycle Thermal Input	93.4 kW	

Table 3 provides a more detailed description of the basic power module internal loss mechanisms and their magnitudes. From the thermal viewpoint, the conduction loss past the regenerator in the heat exchanger module represents an area where further refinement of the hot-end materials could result in a significant increase in system performance. In the power case, displacer losses represent an area for possible power improvements; however, the magnitude of any changes is highly dependent on the specific displacer seal and gas spring design employed.

Table 4: Power Module Mass Breakdown

	Fraction of Total Mass %	
Alternator (Stationary)	27	
		(Alternator 33)
Piston	6	
Displacer	2	
Pressure Vessel (Cold)	16	
Heat Exchanger Modules	29	
		("Hot End" 23)
Pressure Vessel (Hot)	11	
Dynamic Balance Unit	9	

Total Mass of Engine = 154 kg

In conjunction with the goal of high performance at the low temperature ratios, it was also necessary that the power module have a low specific mass. This latter requirement was the primary driver for the high operating pressure and frequency of the reference design which in turn increased the demand for the use of advanced material fabrication techniques. The current design mass breakdown is shown in Table 4. Critical component dimensions and masses are listed in Table 5.

The remaining portion of this technical review describes the various components of the power module and discusses the key design criteria employed with particular emphasis placed on materials and manufacturing issues.

Table 5: Component Characteristics

Power Module	
Overall Length	905 mm
Maximum Diameter	428 mm
Module Mass	154 kg
Power Piston	
Diameter	160 mm
Length	290 mm
Moving Mass	12 kg
Bearing / Seal Gap	1.875×10^{-5} m
Gas Spring Seal Gap	2×10^{-5} m
Displacer	
Diameter	160 mm
Length	200 mm
Moving Mass	4.24 kg
Gas Spring / Bearing Seal Gap	1.25×10^{-5} m
Cylinder Seal Gap	1.875×10^{-5} m
Balance Mass	
Diameter	136 mm
Length	100 mm
Moving Mass	12 kg
Bearing / Seal Gap	1.25×10^{-5} m
Heat Exchanger Module	
Heater Length	120 mm
Passages Size and Number	1.3 mm X 3.8 mm (21)
Regnerator Length	38 mm
Regenerator Frontal Area	0.001 m^2 per module
Regenerator Porosity	0.825
Wire Diameter	2.5×10^{-5} m
Cooler Length	120 mm
Passages Size and Number	1.6 mm X 2.4 mm (18)

MATERIAL / MANUFACTURING ISSUES

During the investigation of the current power module it was evident that materials and manufacturing issues would play a major role in its development. As can be seen in Figure 2, the FPSE and its associated linear alternator employ a number of different materials throughout its structure. In some cases, such as the permanent magnets and special property lamination materials, there are no alternatives from the materials viewpoint.

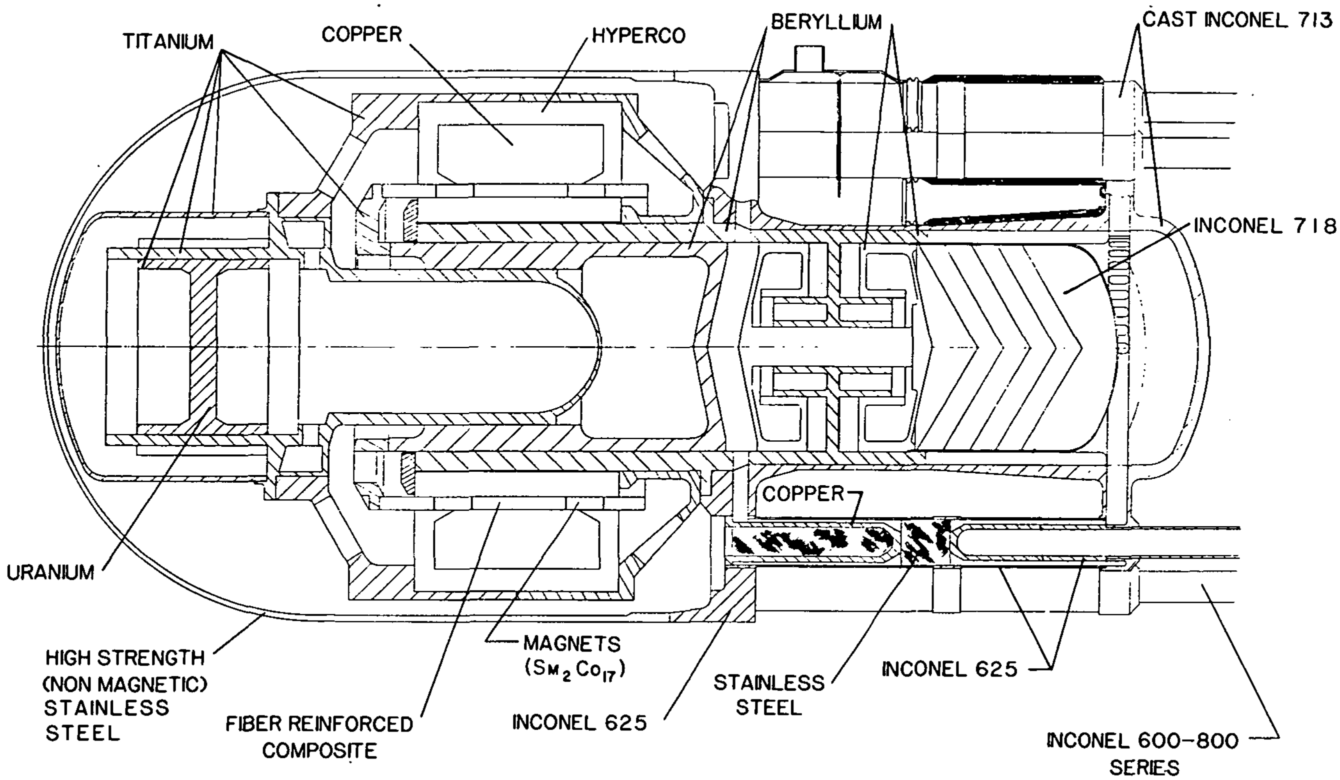

Figure 2: Power Module Materials

The moving components employ beryllium due to its low mass and excellent dimensional stability; however, the use of ceramics, particularly on the power piston, was given serious consideration. The cold-end pressure vessel is primarily fabricated from high strength steel alloys which are attached via welding to the Inconel 625 bulkhead used to support the moving component cylinders. Forward of the stainless steel regenerator section the temperatures rapidly increase, requiring the application of the higher performance nickel-based superalloys for the primary pressure vessel

and heat pipe support structure. This transition between the wrought Inconel 625 and the cast Inconel 713 series material represents the critical fabrication issue in the current design. The joints between the 40 heat pipe modules and the cast structure, while critical, do not carry the loads experienced by the main pressure vessel joint. At present these joints will be formed by a friction welding process that is a well established technology for Inconel 713 components used in high speed equipment (turbo charger rotor systems, for example). However, it is expected that the particular application of this technique on the primary pressure vessel will need further development.

Since the hot-end structure represents the most demanding materials application area, a major portion of the design effort was focused on this section of the power module. Throughout the program the basic criteria, shown in Table 6, were applied to the material selection process for the hot-end components. The criteria were developed from basic engineering design techniques concerned with creep limited structures.

Table 6: Material Design Criteria

System Operating Life ~ 62,000 Hours

Operating Stress Limited to 90% of Stress Producing 1% Creep in 85,000 Hours

Heat Pipe Material Compatible with Na up to ~ 1100 K

Easily Joined by Welding or Brazing

System operating time is approximately seven years at full power and temperature. This has been increased to approximately 85,000 hours (9.7 years) to take into account the effect of the engine cycle pressure oscillations. Since the primary structure is tolerant to some permanent deformation over its design life, a design stress level of 90 percent of the stress to produce 1 percent creep deformation in 85,000 hours was employed. These criteria represent reasonable limits; however, as uses for the power module become more defined, it is possible that different criteria may be required. For example, a "four times design life" criterion has been discussed when the system would be employed with or near manned spacecraft.

During this materials review, it became evident that a number of trade-offs existed concerning system operating life and hot-end temperature. If the specific mass of the system were to remain essentially constant (See Figure 3.), two distinct operating regimes are available. At temperatures below approximately 975 K, the unit could be designed to meet operating life requirements with the well defined superalloys and relatively straight forward manufacturing/fabrication techniques. This would, however, drive the effective cycle temperature ratio to approximately 1.86 if a cold-end temperature of 525 K is retained, reducing overall system efficiency and power output. As shown in Figures 4 and 5, power module performance (for a fixed engine design) begins to rapidly fall off at temperature ratios below 2.0. Note that Figures 4 and 5 were for an earlier version design point with a hot-end temperature of 1080 K, but the current results would be similar.

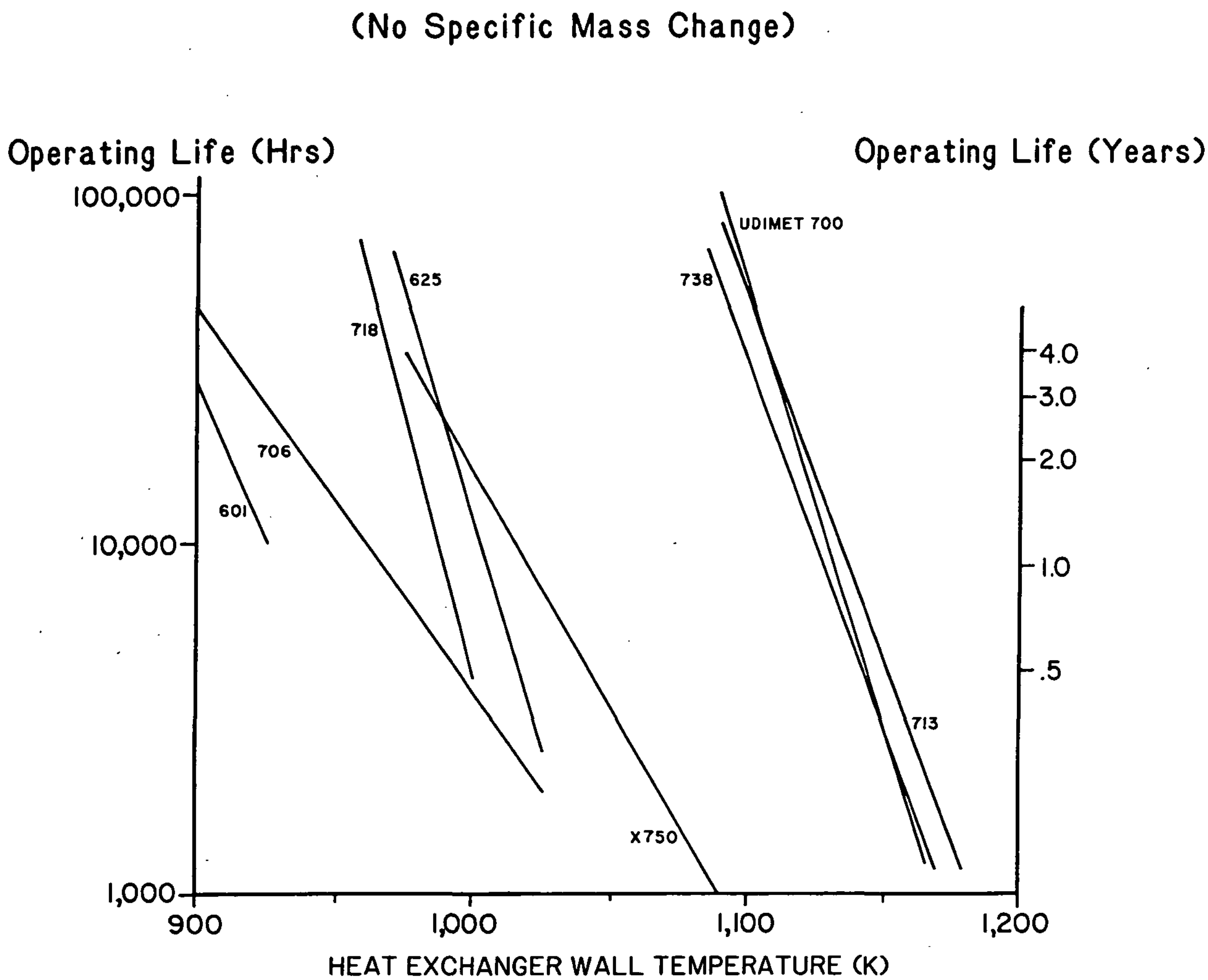

Figure 3: Impact of Materials Selection on Module Operating Life

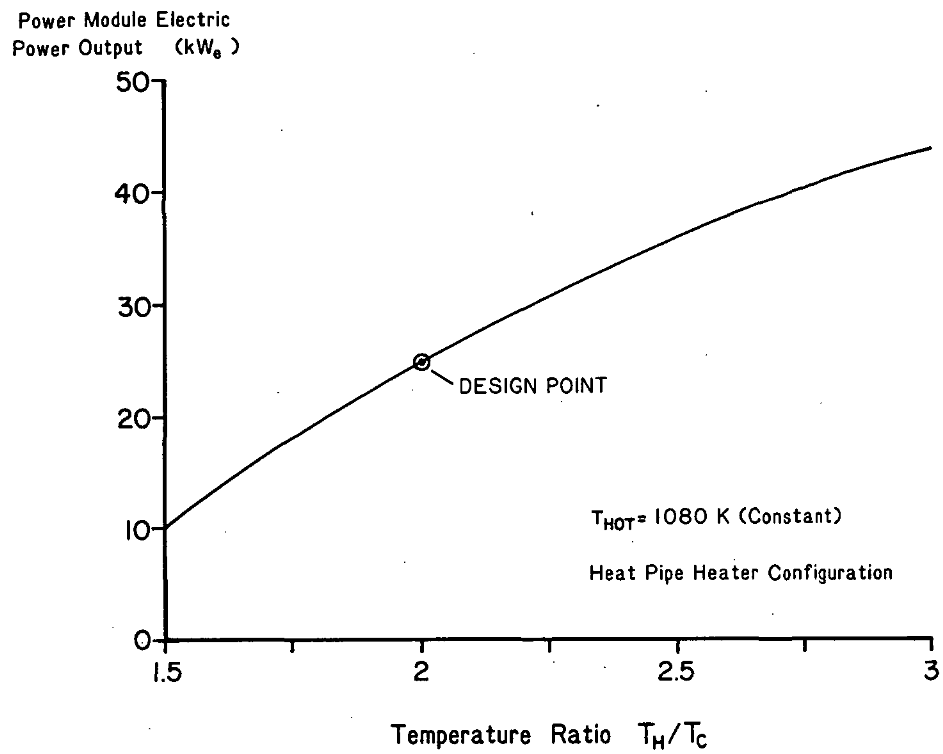

Figure 4: Impact of Cold-End Temperature on Performance (Power)

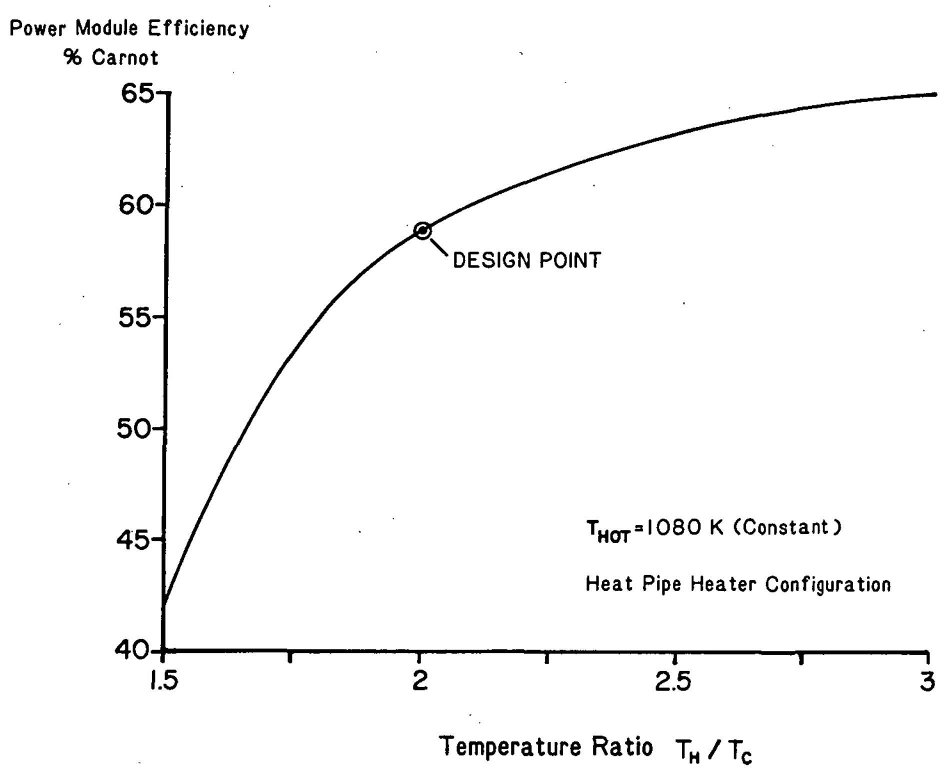

Figure 5: Impact of Cold-End Temperature on Performance (Efficiency)

The less well defined cast superalloys, as indicated in Figure 3, clearly meet the general performance requirements at the expense of more complex issues in the fabrication area.

Another approach investigated was allowing specific mass to vary, as shown in Figure 6. Again, it is clearly evident that the two basic types of superalloys available determine operating temperature.

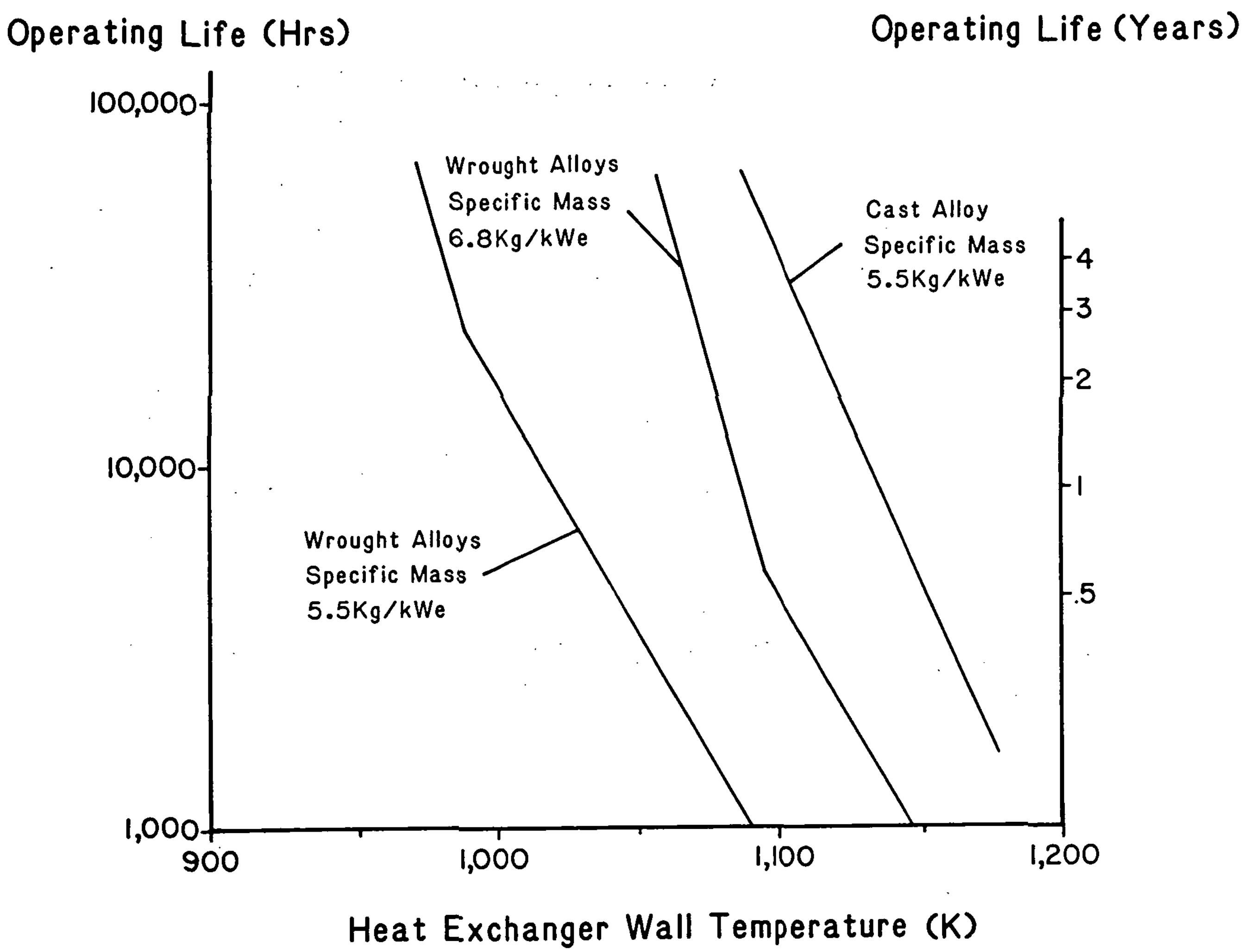

Figure 6: Impact of Increased Specific Mass on Module Operating Life

Irrespective of the particular criteria employed, it can be seen in Figure 7 that the design point of the current unit places severe limitations on the available materials and their useful strength. Early in the program it was evident that the Inconel 713 family represented the only practical material for the highly stressed hot-end pressure vessel with Inconel 625 and Inconel 718 useful for the cooler portions of the structure or the lower stressed, high temperature components.

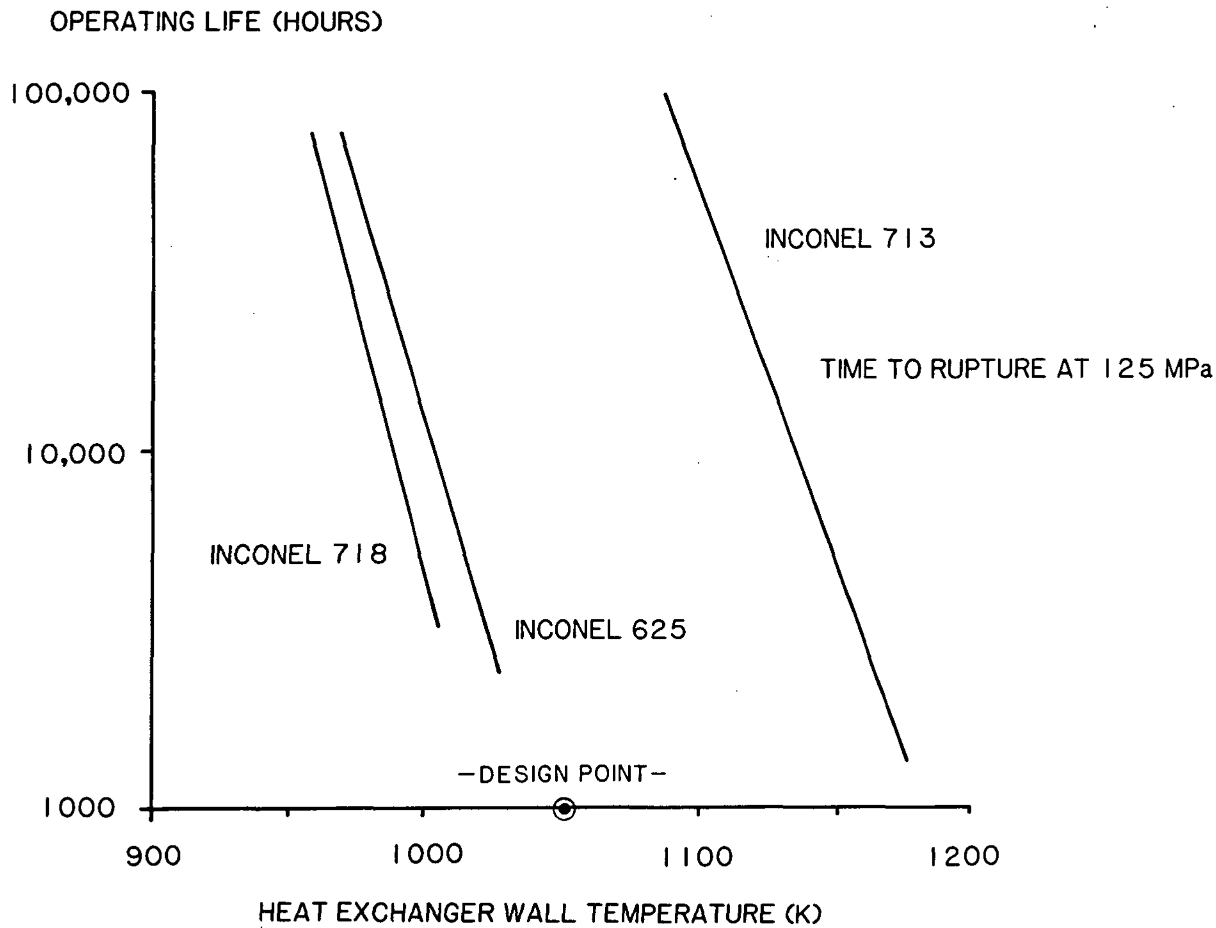

Figure 7: Operating Temperature vs. Life

In the current design (See Figure 8.) this cast material is employed for the displacer cylinder, hot-end manifold and heat pipe support, with a joint to the Inconel 625 occurring in a cooler portion of the displacer cylinder. The basic assembly process is described in Table 7, clearly pointing out the critical joining requirements. This assembly process also plays a role in the selection of some of the cold-end heat exchanger materials since a number of high temperature brazing / welding / heat treatment operations will be required for the entire heat exchanger module during fabrication.

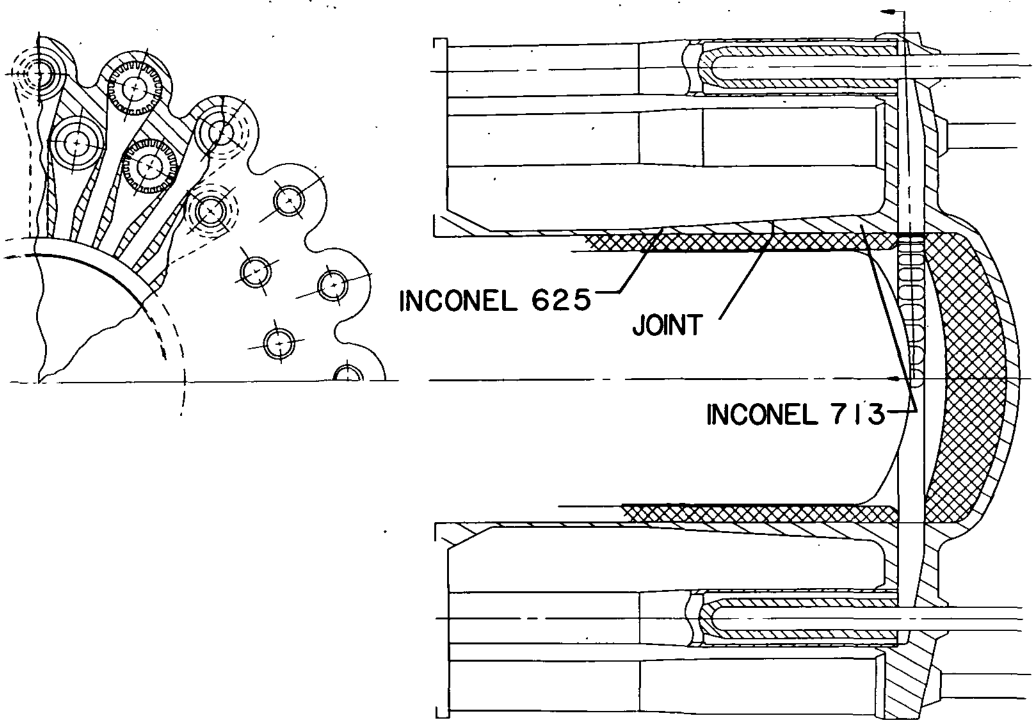

Figure 8: Hot-End Assembly

Table 7: High Temperature Component Assembly Process

Cast Inconel 713 Hot-End Pressure Vessel and Manifold

- Friction Weld to "Cold-End" Inconel 625 Pressure Vessel and Install Inconel Heater Inserts (See Figure 15 for Details of Inserts.)
- Heat Exchanger Modules Inserted Through Cold-End Manifold and Welded
- Hot-End Heat Exchanger Module Brazed to Heater Insert

The development of a ground-based test engine employing an Inconel 625 hot end rather than Inconel 713 was also investigated during this program. The results indicate that the design life can be attained at reduced hot-end temperatures (about 1000 K) or the full temperature maintained for about 6,000 hours to 10,000 hours of engine operation.

Another critical materials selection case is that of the stationary structure supporting the linear alternator assembly. Ideally, this material should be of extremely high electrical resistivity to minimize the amount of induced currents within the components caused by the rapidly varying magnetic field. If fatigue life criteria can be met, the use of nonmetallic fiber-reinforced composites would allow an extremely stiff, lightweight structure to be fabricated with low parasitic power losses.

Finally, the hot-end heat exchanger represents an important materials issue. The inner wall of the heat pipe must be compatible with sodium over its 60,000 hours life. Since the heat pipe wall on the condenser end also forms the fins and gas passages for the helium working fluid, it must have the required strength to withstand the stresses in this area. However, the wall thickness must be minimized to maintain low temperature drops across the wall.

HEAT EXCHANGER MODULES

The current design employs a unique modular heat exchanger assembly to carry out the critical heat input and heat rejection portion of the Stirling cycle. Previous designs have employed variations of the shell-and-tube heat exchanger configuration with reasonable success. However, to attain the high performance required in the proposed FPSE, a configuration requiring 2,000 to 5,000 small (approximately 1 mm internal diameter) tubes would be required for both the heater and cooler. Since each tube must be joined to the pressure vessel structure at both ends, approximately 8,000 to 20,000 brazed joints must be produced, all of which must seal the extremely high pressure helium working fluid with about half operating at the hot-end temperature. While such a configuration is not impossible to manufacture, it was decided early in the current effort to investigate alternative configurations that reduced mechanical complexity, minimized the number of brazed joints and allowed as much "checking" as possible before installation into the engine itself.

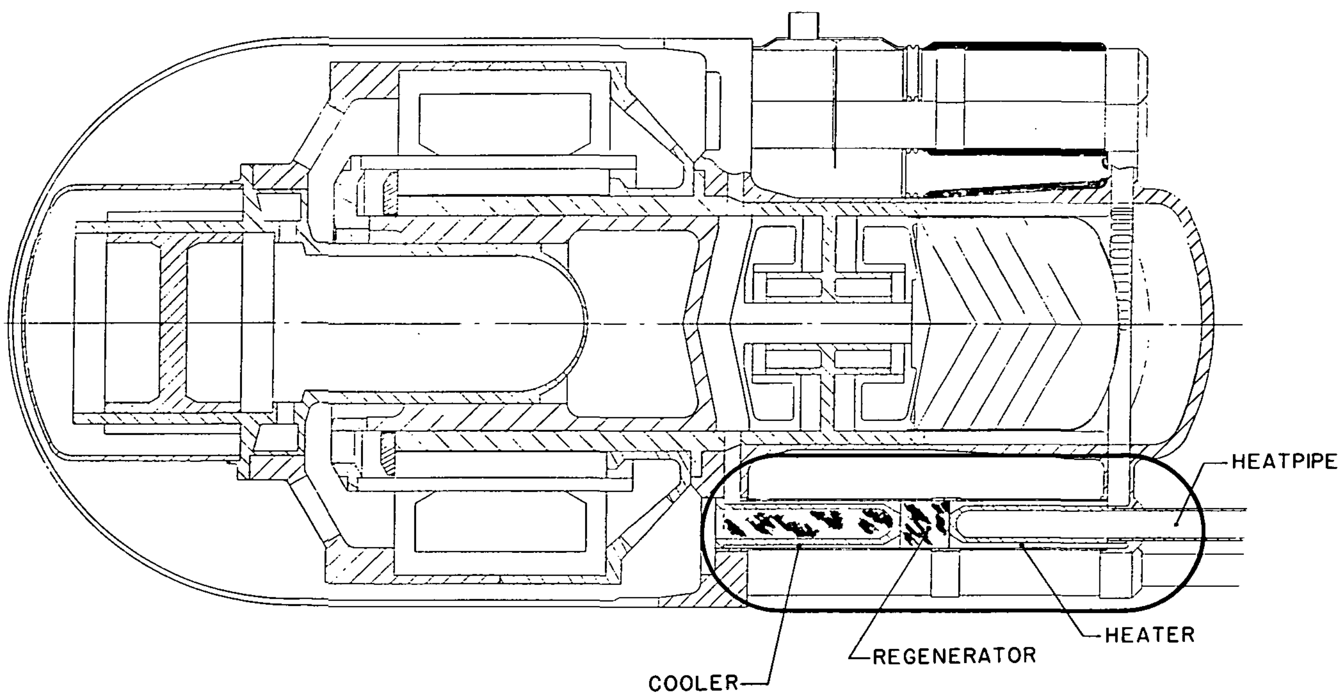

Figure 9: Power Module Heat Exchanger Assembly

The basic configuration developed is shown in Figure 9 in which 40 individual modules are employed. Each module contains a heater portion, with its associated heat pipe, a regenerator element and a cooler or heat rejection portion.

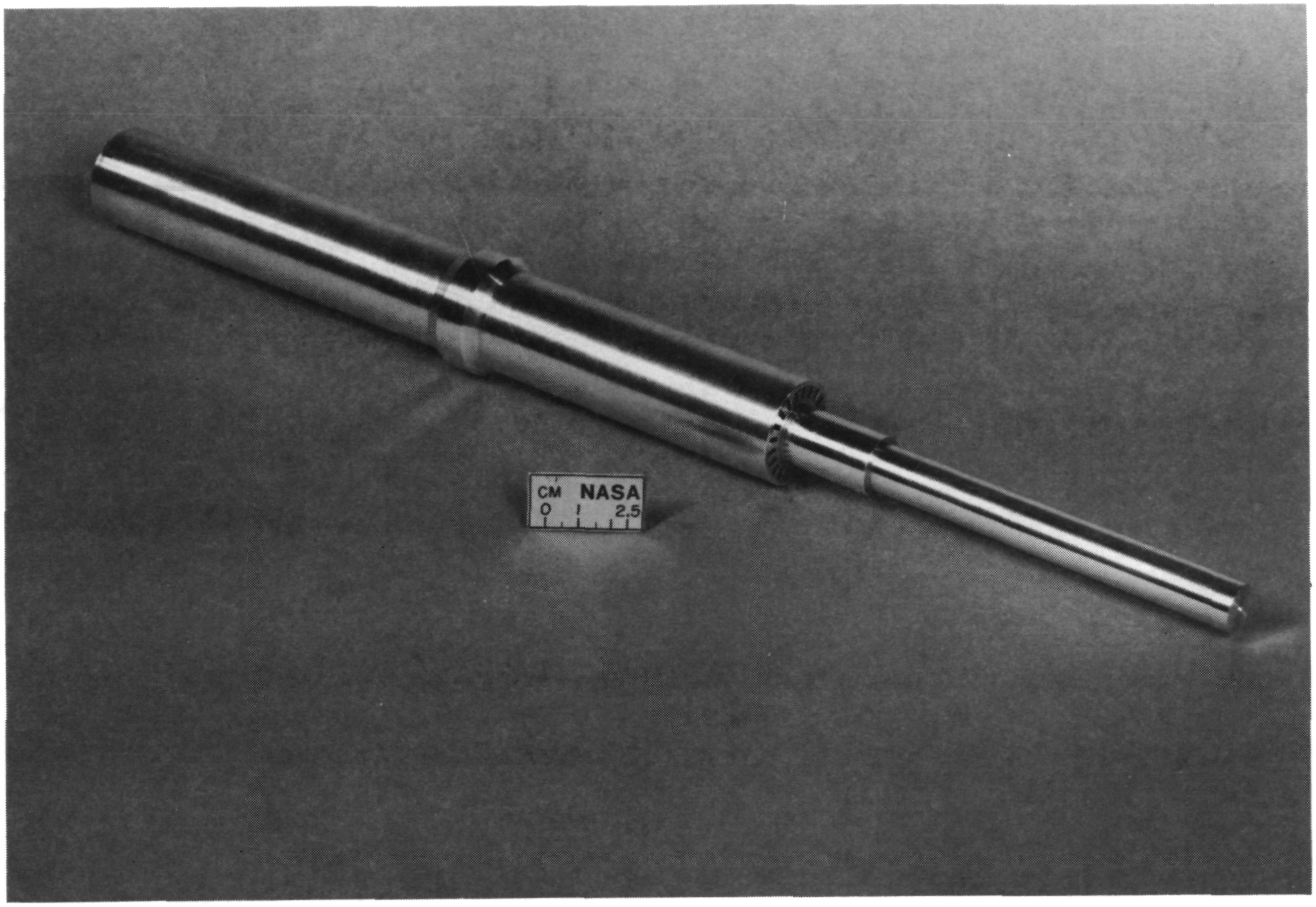

Figure 10: Individual Heat Exchanger Module

Figure 10 is a photograph of a full-scale mock-up of a heat exchanger module. Overall length (excluding heat pipe) is about 280 mm with a maximum diameter of 40 mm. These elements represent all the key components of the heat exchanger loop of a FPSE and as such can be tested on a Stirling cycle simulator to check flow and heat transfer characteristics.

Within the module itself are the helium flow passages which essentially represent the "tubes" of the conventional shell-and-tube heat exchanger configuration. The actual number of passages and their specific geometry was selected by using Stirling cycle optimization codes. At present 21 passages are employed on the heater and 18 on the cooler for a total of 840 and 720 passages, respectively. Since 40 modules are used, only 80 brazed joints are required to the pressure vessel itself, a radical reduction in comparison to conventional designs.

Figure 11 shows the components making up the module. From the left there are: the cooler insert, the regenerator "can" which contains a sintered woven stainless steel screen assembly of 25 micron diameter wire, and the heater-heat pipe hot end.

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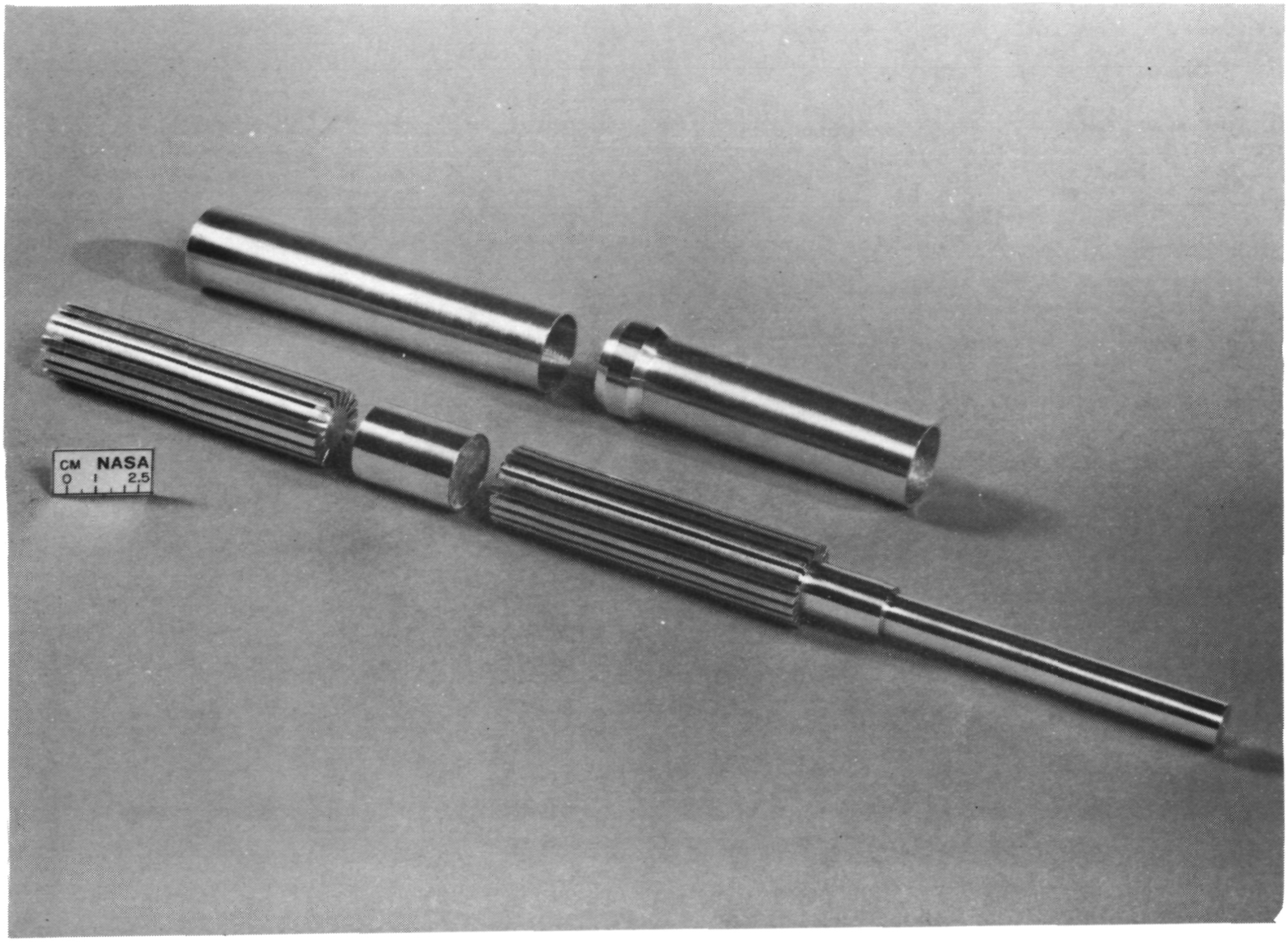


Figure 11: Heat Exchanger Module Internal Components

The finned portion of the heater is integral with the condenser portion of the heat pipe which can be checked out prior to installation into the module. The exterior close-out can be either a two-piece assembly, as shown, or one continuous tube.

As previously pointed out this configuration has a number of advantages over a conventional shell-and-tube heat exchanger arrangement (See Table 8.) There are, however, some disadvantages which, it is believed, can be minimized in further refinement of this concept.

Table 8: Heat Exchanger Modules

Advantages

- Individual Module Contains All Heat Exchangers
- Radical Reduction in Critical Joints
- Complete Testing of Module Prior to Engine Assembly
- Straight Forward Application of Heat Pipes (Hot End)
- Freedom in Internal Passage Geometry Selection

Disadvantages

- Some Reduction in Engine Temperature Ratio vs. "Total" System Temperature Ratio

The primary issue is the mixing of heat transport systems, i.e. heat-pipe heater and pumped-loop cooler. This configuration results in a compromise in module geometry since the heat transfer occurs on the inside of the heater and the exterior of the cooler. In an attempt to minimize heat pipe diameter so as to reduce loading on the structure, the exterior diameter of the cooler could become so small that the pumped-loop surface area would be too small and require higher pumping power. If a taper is placed in the module, i.e. larger diameter cooler than heater, the manifolds on the hot end increase in size along with the maximum diameter of the system and require a nonconstant cross section regenerator.

HOT-END HEAT TRANSPORT SYSTEM

During the initial phase of the current hot-end design effort it became evident that the pumped liquid-metal loop posed a number of mechanical design problems and thermal performance penalties in comparison to a heat pipe configuration. In its basic form the heat pipe system "decoupled" the thermal energy source from the FPSE heater assembly as shown in Figure 12. This allows an individual heat exchanger module to be served by a single heat pipe which can be assembled and tested prior to its inclusion into a complete hot-end assembly with its cast superalloy pressure vessel and support structure.

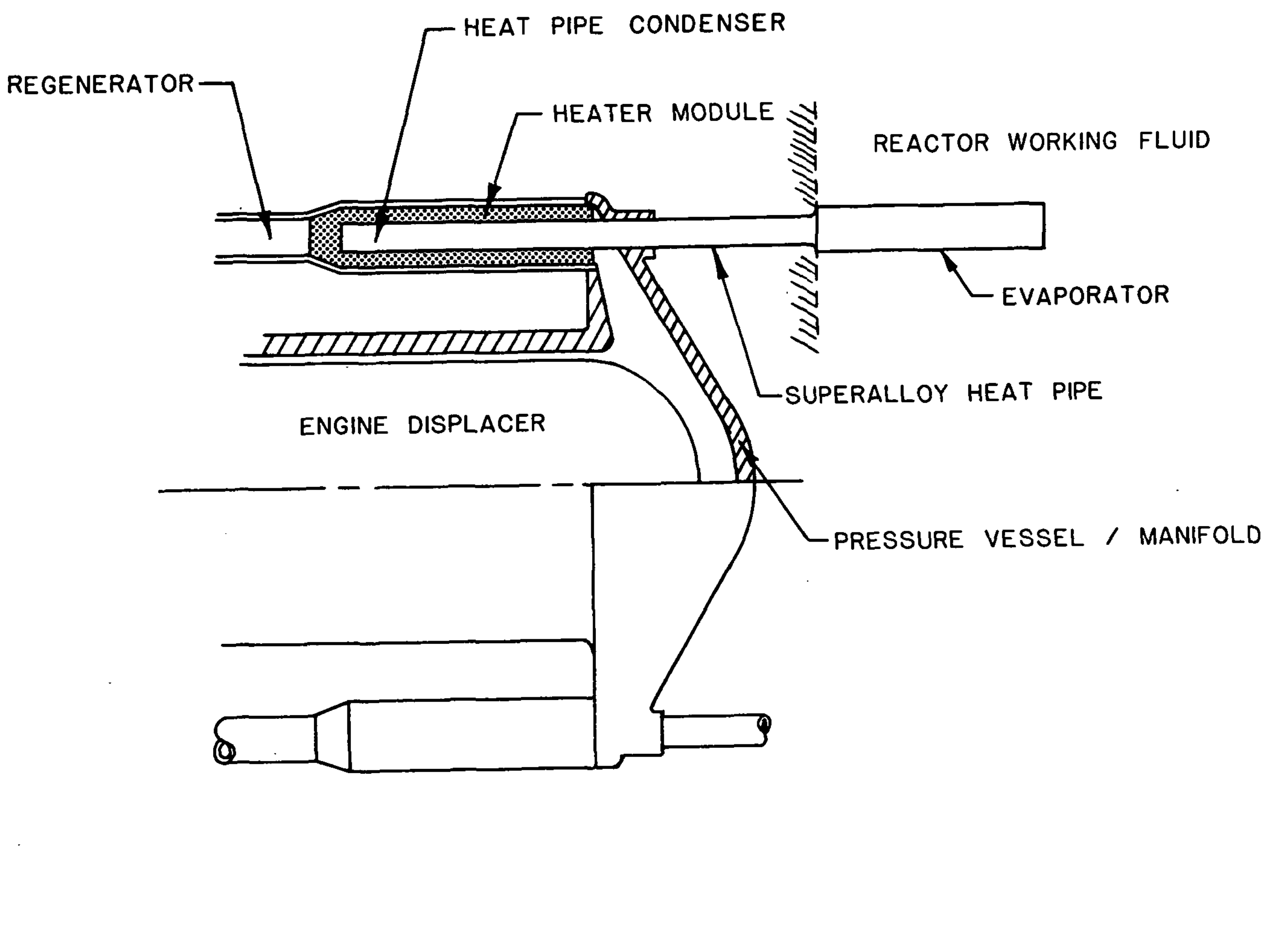


Figure 12: Heater Module — Heat Pipe Arrangement

During the design process the number of individual modules making up the heat transport system was varied between approximately 30 and 75. Selection of the final number was based on a compromise of heat pipe diameter and capacity as well as the geometric packing of the modules in an annular set of rings around the hot-end pressure vessel. This packing played a major role in determining the overall diameter of the system as well as the size and volume of the manifolds connecting each module to the expansion space. The final configuration employed 40 modules in two annular rings of 20 modules, resulting in the heat pipe requirements shown in Table 9. Sunpower employed Thermacore, Inc., to review the basic heat pipe criteria in this process to assure that the heat pipe represented a reasonably conservative design point.

Table 9: Hot-End Heat Exchanger

Design Operating Point

Total Heat Input	93 kW
Heat Pipe Design Point	2.3 + kW
Working Fluid	Sodium
Operating Temperature Regime	1050 + K
Pipe Dimensions	
•Inside Diameter (Wall)	22.4 mm
•Condenser Length	115 mm
•Average Heat Flux	~ 30W / cm^2

The hot-end modules are connected to the engine expansion space via 40 manifold passages which are integral with the cast hot-end pressure vessel. These passages, which are semicircular in cross section, represent a major portion of the hot-end dead volume and surface area, and as such can impact overall performance. Pressure drops through these passages have been minimized with the heat exchanger module itself representing the major pressure drop so as to assure uniform flow to each module. The actual heater element is approximately 120 mm in length and 38 mm in diameter with each module having 21 flow passages for the FPSE helium working fluid. (See Figure 13.)

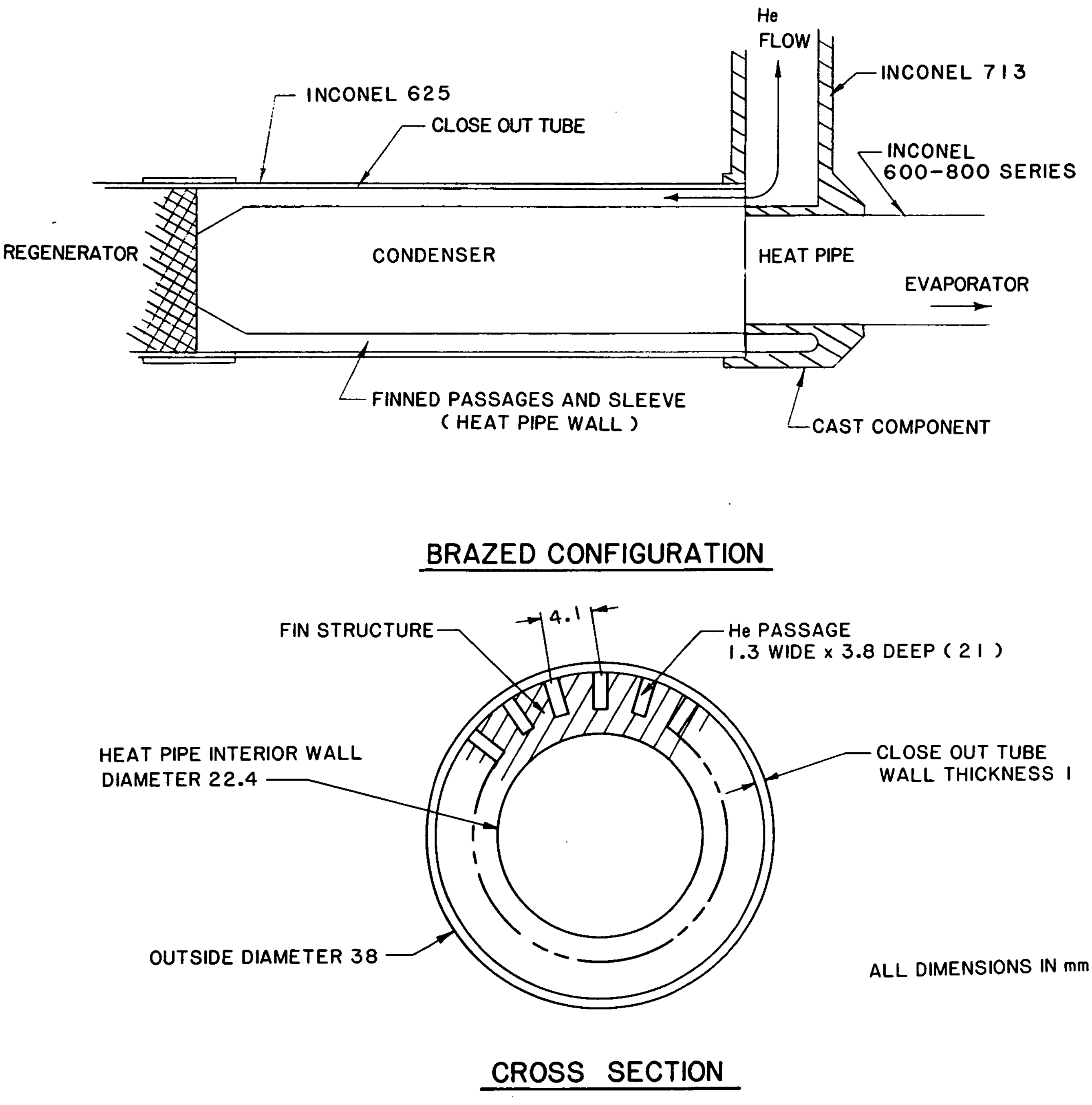


Figure 13: Hot-End Heat Exchanger

Inconel 625 and Inconel 713 are used throughout this structure due to their high temperature capabilities. The passages are closed out by brazing a sleeve over the finned heat pipe exterior, which extends down the entire heat exchanger module length. A possible alternative configuration would employ passages drilled into a thick-walled heat pipe shell, as shown in Figure 14. This technique eliminates the need for an external shell to close out the helium flow passages with its corresponding braze operation. Various machining operations could be employed to provide either round or rectangular passages, however, some geometry constraints are expected on passage configuration due to the relatively high length-to-diameter ratios employed.

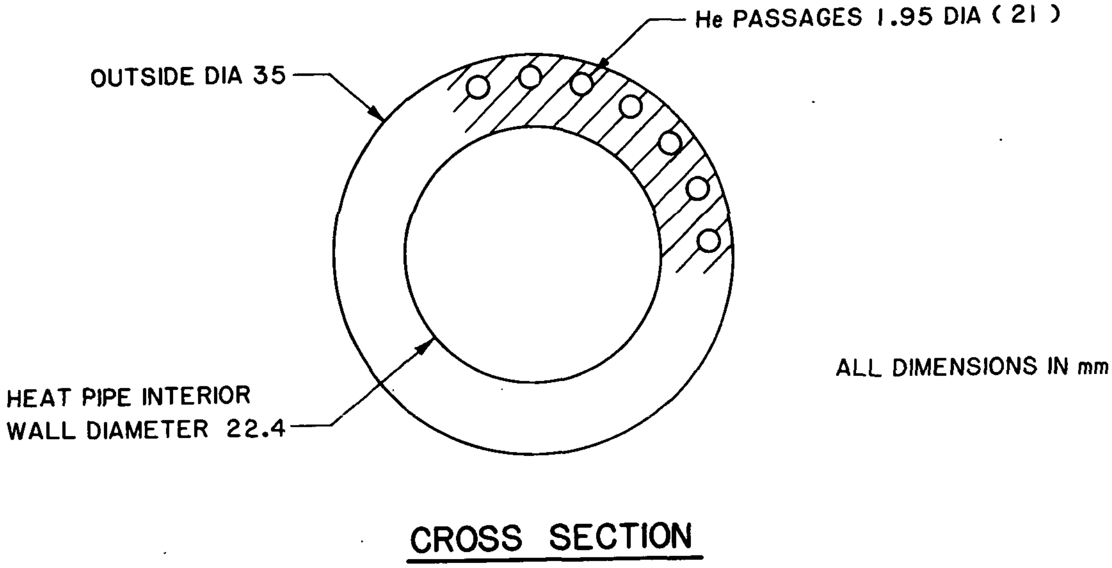


Figure 14: Drilled Passage Configuration

In the current design extremely conservative criteria have been employed to determine the necessary wall thickness from the heat pipe inner wall to the base of the helium flow passages. This thickness is important since the temperature drop from the heat pipe inner wall to the helium wall is directly proportional to this value for a given heat input. If the wall is excessively thick, the heat pipe will have to operate well above the helium wall temperature; if it is too thin, structural failure will occur. The current design requires approximately 25°C temperature drop, but refined structural analysis may allow this value to be reduced.

While not specifically a part of the hot-end heat transport system, the structural joint between the individual hot-end modules and the cast Inconel 713 pressure vessel, shown in Figure 15, represents a critical manufacturing issue.

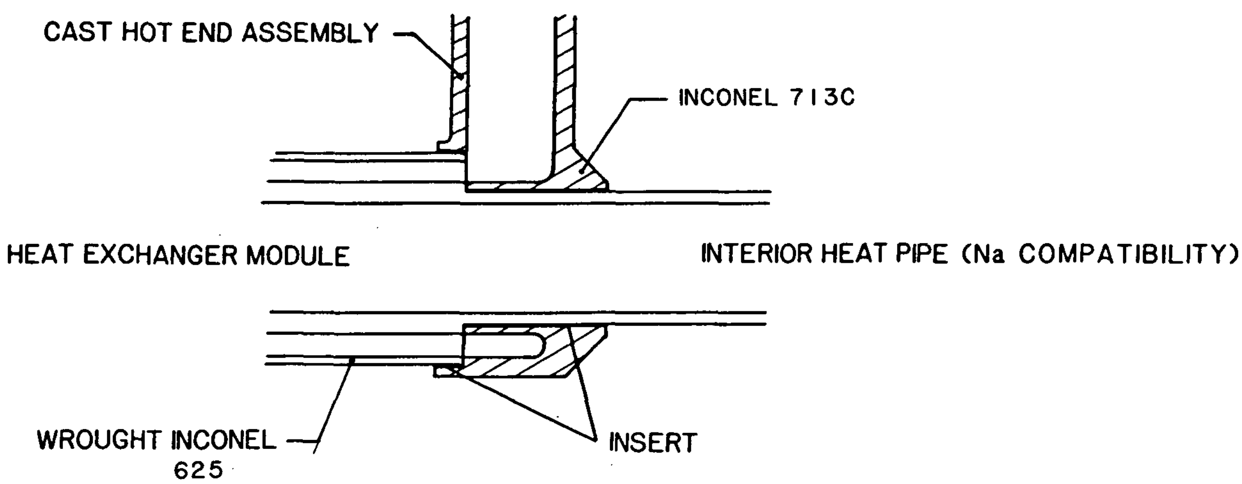


Figure 15: Critical Materials / Manufacturing Issue

As currently configured, Inconel 625 inserts are friction welded into the cast structure, followed by the brazing of the Inconel 625 heat exchanger module. The joint configuration must be reviewed from the viewpoint of thermally induced stresses occurring during the start or shut down of the system.

In conclusion, the hot-end heat transport system represents a considerable improvement of previously investigated pumped-loop arrangements, however, a number of materials / fabrication issues must be addressed in conjunction with a refined structural analysis as indicated in Table 10.

Table 10: Hot-End Heat Transport

Conclusions

- Materials Issues Play a Major Role on Heat Pipe Maximum Operating Temperature Requirement

- Refined Structural Analysis Required
  - — Joint Configuration
  - — Module Failure Modes

- Brazed Configuration Allows Wider Selection of Passage Geometry and Possibly Better Thermal Performance

- Drilled Passage Configuration Eliminates a Number of Fabrication Steps, but Places Some Restrictions on Passage Geometry

COLD-END HEAT TRANSPORT SYSTEM

The cold-end heat transport system retains the pumped-loop configuration due to its simplicity in interfacing with an external heat rejection (radiator) system and due to the nonavailability of heat pipe working fluids that meet all system requirements. Since 40 heat exchanger modules extend from the hot end, 40 are also employed in the cold-end system; however, in this case the thermal energy is removed from the exterior of the modules by flowing a coolant over the outer surface. The basic configuration of the heat rejection system is shown in Figure 16 and a schematic of the coolant's flow path in Figure 17.

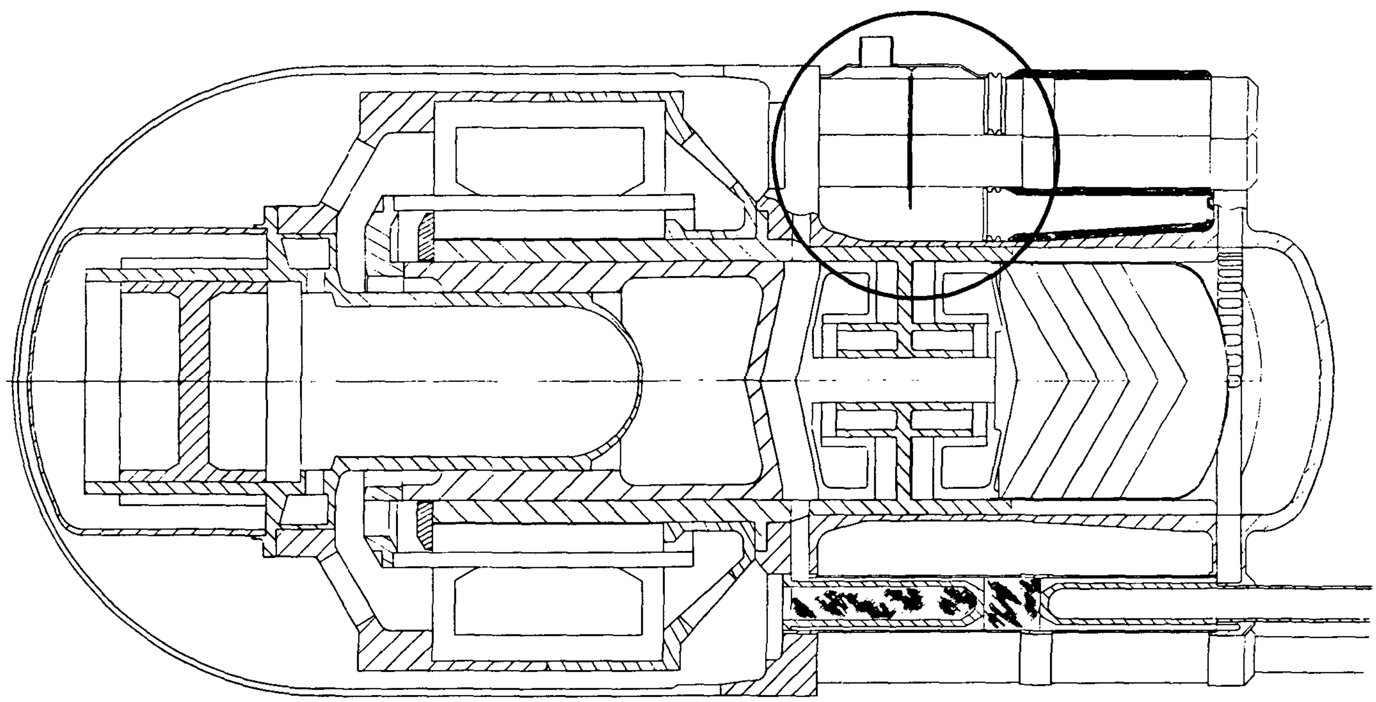


Figure 16: Cold-End Heat Transport System

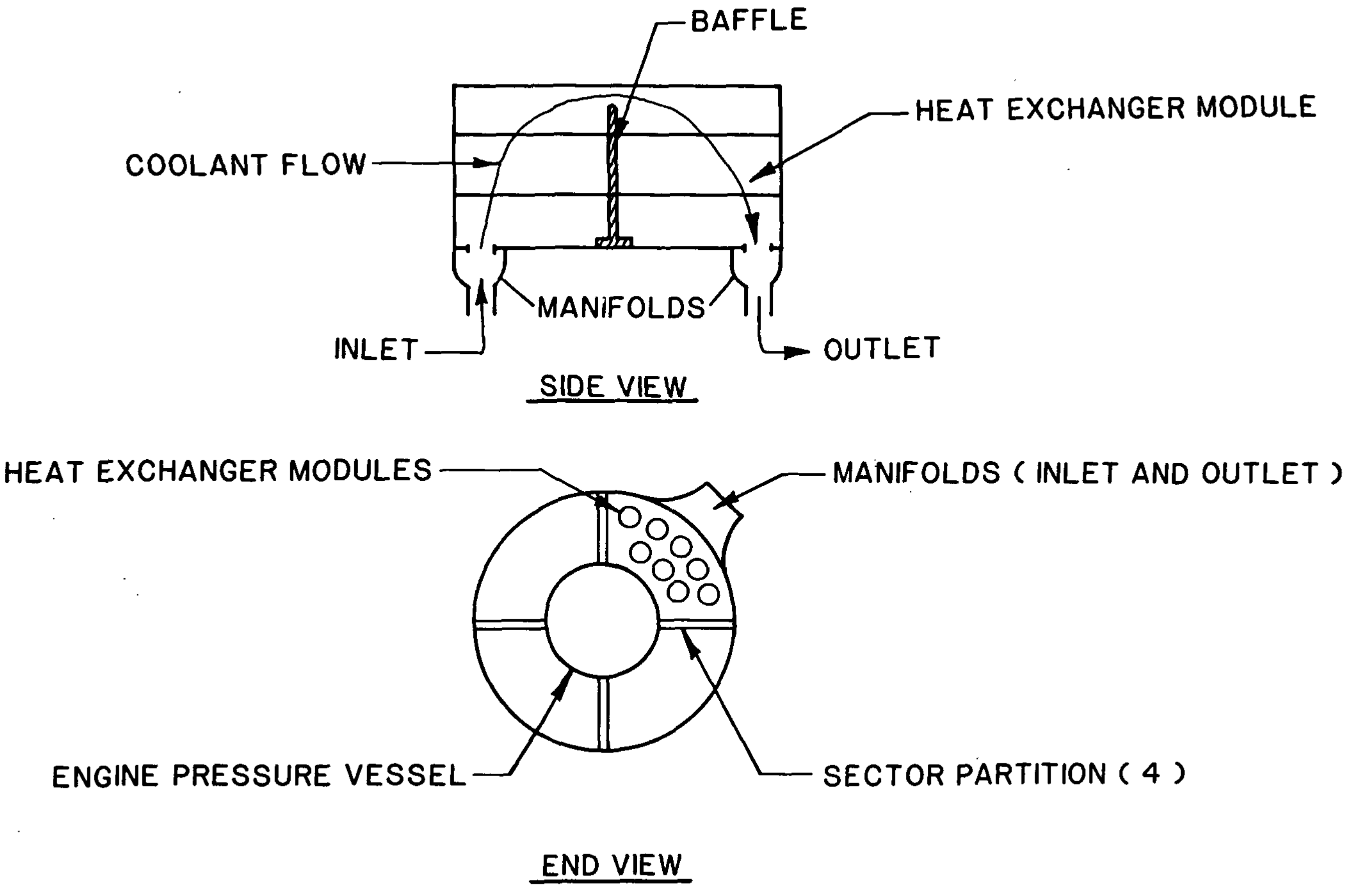
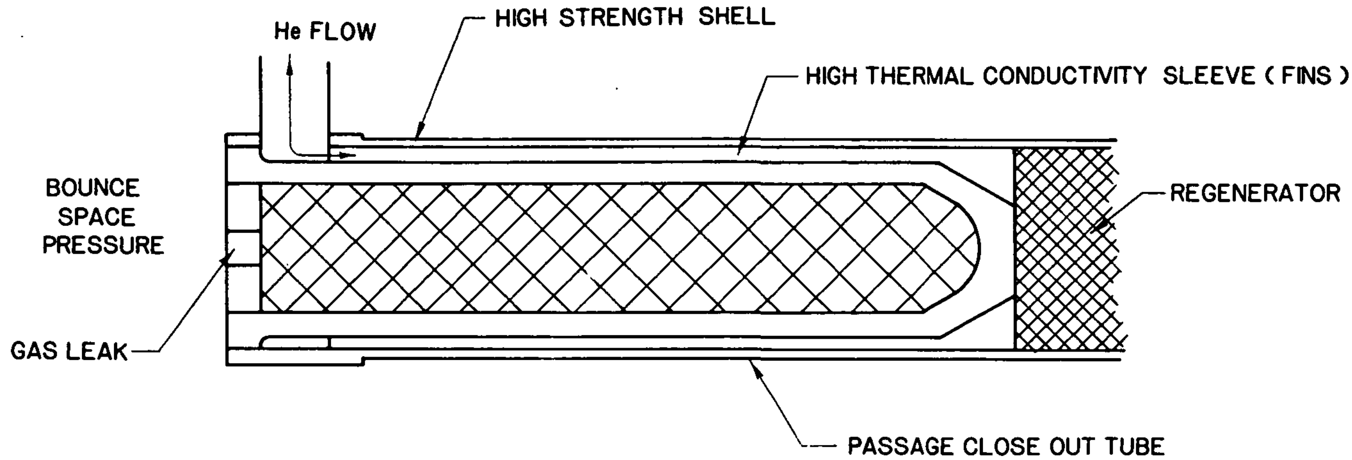


Figure 17: Cooler Working Fluid Flow Arrangement

The purpose of partitioning the cold end into four sections is to assure more uniform flow of the coolant over the heat exchanger exteriors.

The individual modules, shown in Figure 18, are approximately 120 mm in length and 40 mm in external diameter with 18 helium flow passages provided in each module. The center of the module is stuffed with a low density metal fiber, similar to the material in the regenerator, which is vented via a capillary leak to the engine space where the linear alternator is housed. The pressure in this volume is then the mean engine pressure allowing the wall between the base of the helium passage and this volume to be quite thin since it must only carry the pressure oscillation of the engine cycle (about 20 bar) rather than the full engine pressure as occurs between the hot end of the module and the heat pipe.



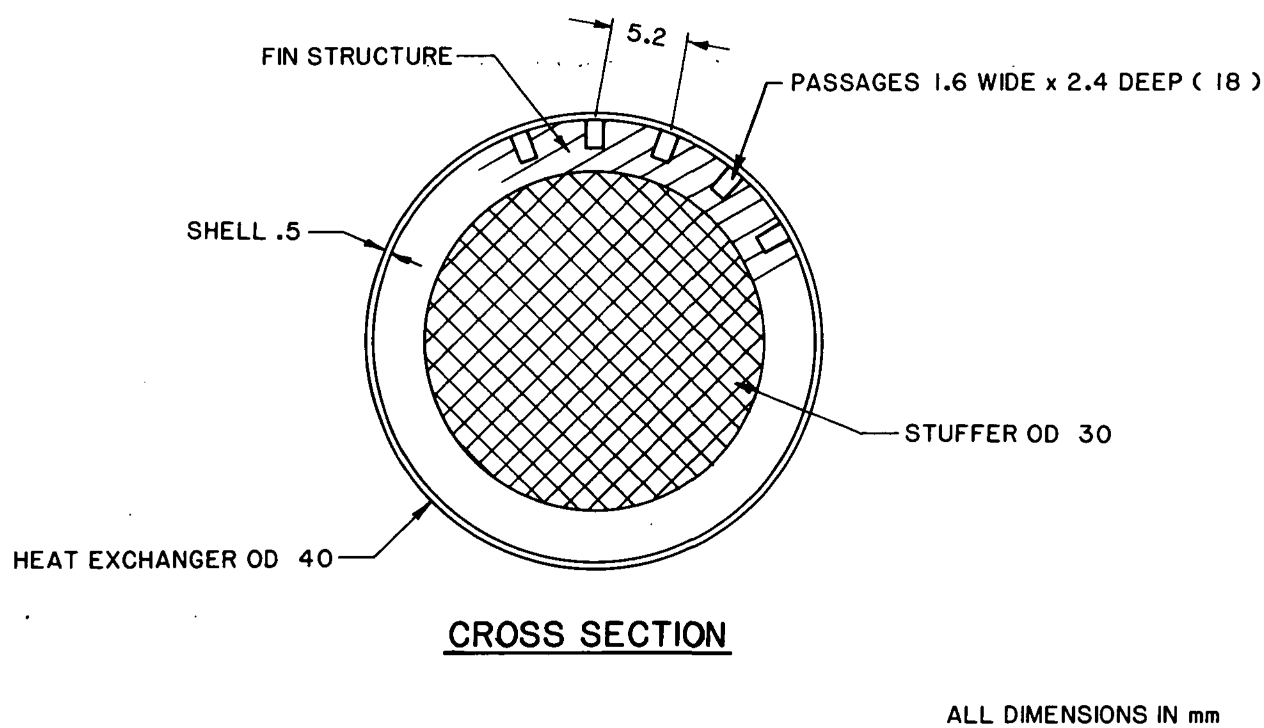


Figure 18: Cold-End Heat Exchanger

Due to the lower operating temperature (approximately 525 K), materials and manufacturing issues are less critical than on the hot end. In the current design OFHC copper is employed for the "fins" of the cooler insert to improve effectiveness; this requires a dissimilar joint with the Inconel 625 tube close-out. For the temperatures normally encountered during operation, this joint is well within current brazing technology; however, some questions exist on the temperatures this area would encounter during fabrication since a number of much higher temperature joints / heat treatments must occur on the hot end. Should these processes exceed the useful OFHC / Inconel 625 capability the OFHC can be replaced with a Nickel 270 alloy with a minor decrease in heat exchanger effectiveness.

The operating point of the heat rejection system is shown in Table 11. Throughout this program the coolant working fluid has been assumed to be a sodium-potassium mixture.

Table 11: Cold-End Heat Exchanger

Operating Conditions

— NaK Inlet Temperature	497 K
— NaK Outlet Temperature	517 K
— Flow Rate	3.5 kg / sec
— Fluid Pumping Power	~ 50 W
— Two Passes Cross Flow	
— Effectiveness	~ 0.80

Due to the complex baffle and manifold shapes required to force the fluid over the exterior of the cooler modules and the potential for leaks with this arrangement, a modification was investigated where the cooling fluid was pumped inside the cooler modules, as shown in Figure 19. While requiring somewhat higher pumping power, this configuration eliminates the above problems at the expense of a pressure vessel joint at the point where the coolant enters the module. Another advantage of the scheme is that the coolant passages could be optimized for a specific cooling fluid whereas the current configuration is limited in this area.

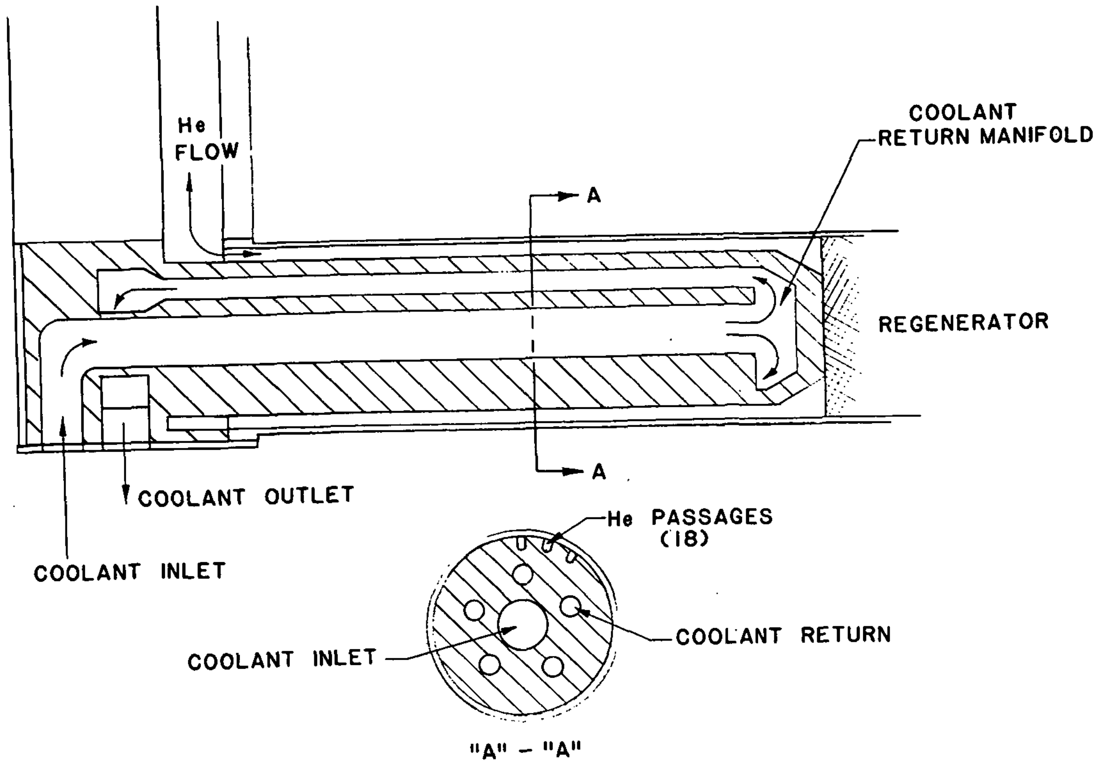


Figure 19: Alternative Cooler Configuration (Pumped Loop)

An investigation was also performed on various heat pipe configurations for the cold end. These may be considered if an appropriate working fluid can meet all the necessary system requirements. Heat pipe configurations considered included an arrangement similar to that employed on the hot-end assembly with the heat pipe located in the central core of the module (See Figure 20.).

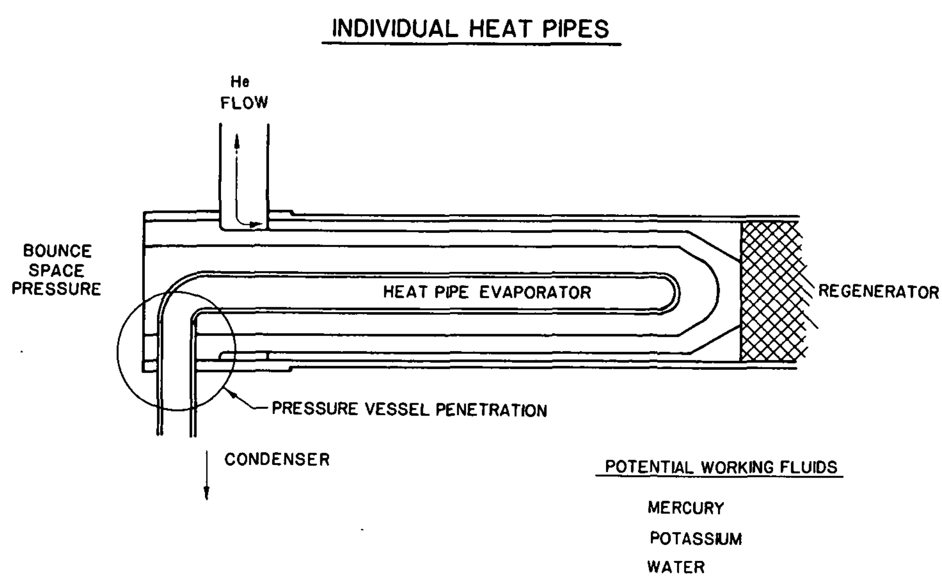


Figure 20: Cold-End Heat Exchanger with Heat Pipes

A critical portion of this concept is the development of an arrangement to allow the heat pipe to penetrate the pressure vessel after the module itself has been installed. If the heat pipe could be fabricated in two parts and "joined" in the area indicated with a circle, this configuration could be integrated into the design. However, technical questions exist concerning the impact of such a joint on the heat pipe wick structure and the ability to adequately test (or repair) the resulting heat pipe. Working fluids investigated included mercury and potassium, both of which can easily handle the heat flux levels encountered in the current design. Due to its inherent simplicity, the heat pipe-based cold-end heat transport system should be aggressively investigated in any further development of this engine concept.

In the case where no auxiliary cooling system is employed to remove the waste heat from the power module rear spaces and linear alternator, the cold-end heat transport system provides this function. The added 3 kW (approximately 2 kW from the alternator and the remaining 1 kW from various internal losses) has essentially no impact on the basic power balance of the cold-end heat exchanger system; however, the inlet temperature of the coolant (about 500 K) will play a major role in determining linear alternator temperature. The current design will require approximately 20°C temperature difference to transfer the alternator waste heat to the coolant coils within the alternator space via circulated helium. This strongly suggests that some alternator components will be operating well in excess of 530 K if this type of heat rejection system configuration is employed.

The heat rejection requirements for the situation where the alternator components and bounce space are maintained at approximately 525 K and the compression space temperature increased are shown in Figure 21. The analysis was based on minimizing the losses due to the convective/conductive heat flow from the hotter spaces and due to the flow of helium working fluid required to center the power piston.

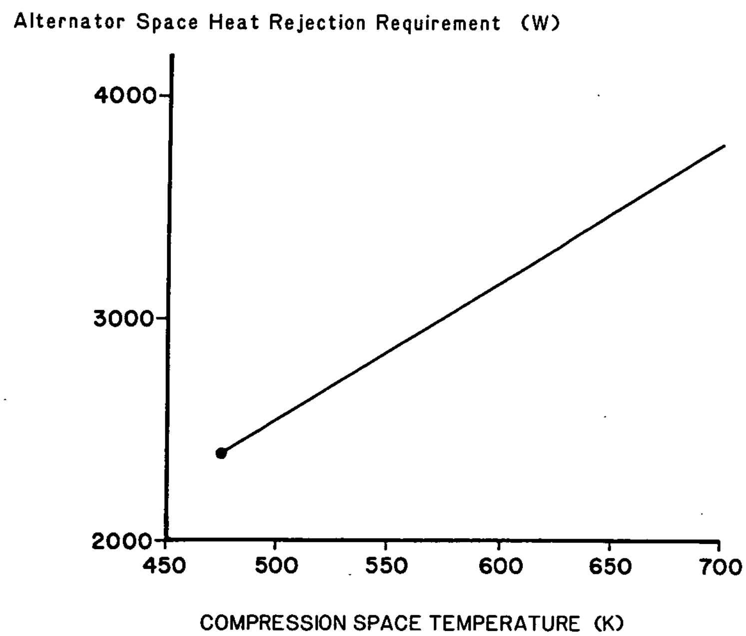


Figure 21: Alternator Cooling Requirements

The use of an independent cooling system, as shown in Figure 22, allows the alternator components to operate at lower temperatures; however, the temperature distributions caused within the structure (particularly the power piston and its cylinder) could lead to differential thermal expansion in the critical bearing and seal area. Such an auxiliary system could employ a simple pumped loop where the cooling fluid is passed through a heat exchanger in the rear space. Due to the spinning of the power piston, which is necessary to provide the hydrodynamic bearing, it is possible to add a "fan" to the piston structure to cause a forced flow of the helium over the alternator components and through the internal heat exchanger. A heat pipe-based heat rejection system was also considered. In either case a number of pressure vessel penetrations would be required which would have to be carefully designed so as to minimize the potential of helium leakage.

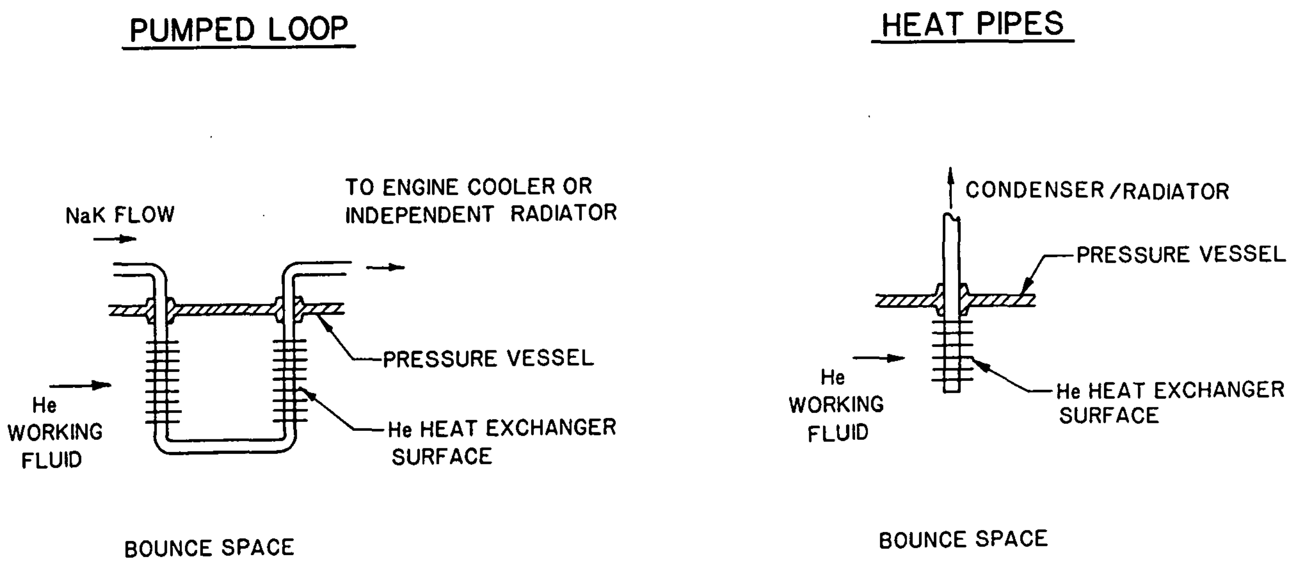


Figure 22: Auxiliary Heat Rejection

POWER PISTON AND DISPLACER

The power piston and displacer are the moving components of a FPSE and as such have received considerable attention since they represent the only components that are subject to wear. In past designs either noncontact pumped hydrostatic bearings or extremely hard-wear pairs (such as ceramics) have been considered for long life FPSEs. For the current design it has been assumed that some form of noncontact bearing / seal must be provided since even hard surfaces would experience excessive wear over the long operating times considered. Due to the complexity of a pumped hydrostatic bearing, Sunpower proposed the use of a "spinning" or hydrodynamic bearing/seal for the piston and displacer. In the proposed scheme, depicted in Figure 23, aerodynamic forces cause both the piston and displacer to spin about their horizontal axis, inducing a gas flow in the bearing/seal gap. These forces are caused by the naturally occurring flow of the helium working gas during the Stirling cycle. As the working gas flows into and out of the compression space, it interacts with turbine assemblies mounted on the piston and displacer, inducing a spinning motion. This concept was tested on the displacer in a Sunpower test rig as part of the work reported in reference 2.

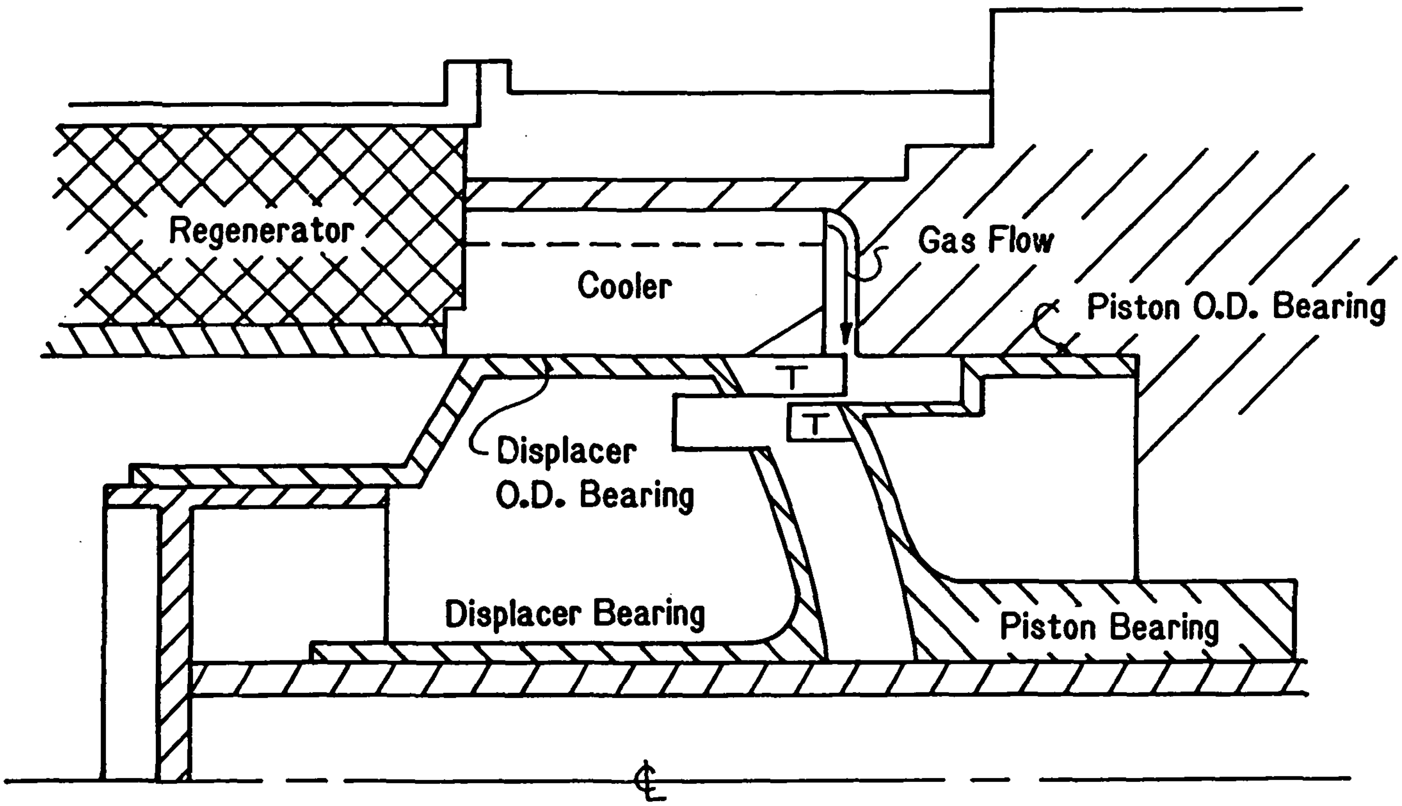


Figure 23: Spin Bearing Arrangement in Stirling Engine

Some questions do exist as to the ability of these aerodynamic forces to spin the piston due to the "resistance" of the nonuniform magnetic field created by the stationary laminations and the permanent magnetic plunger. Should the aerodynamic force not provide enough start-up torque, an electric motor can be used to spin the piston and then shut down when sufficient speed is reached. A hard coating on the beryllium piston and its cylinder is provided since some contact between the piston and cylinder would be expected during start up because the electric spin motor would be powered off the alternator. An electric spin motor is used on the moving component of the balance system since no gas flow occurs past the seals to induce rotation via a turbine.

The power piston without the magnet plunger is a simple cylindrical structure, 160 mm in diameter and approximately 280 mm long with a mass of 8 kg. The exterior cylindrical surface provides a seal between the engine compression space and the alternator space as well as the primary bearing for the piston. Nominal radial gap between the cylinder and piston is 18.8 microns. Concentric with this outer seal is an inner running surface that forms a gas spring within the piston which is required to properly tune the engine dynamics. This surface serves as a combined seal and bearing with a radial gap of 20 microns. (See Figure 24.) The forward portion of the balance piston's gas spring housing serves as the stationary component of the piston gas spring seal. Centering of the power piston is carried out via a controlled leak from the gas spring to the bounce space.

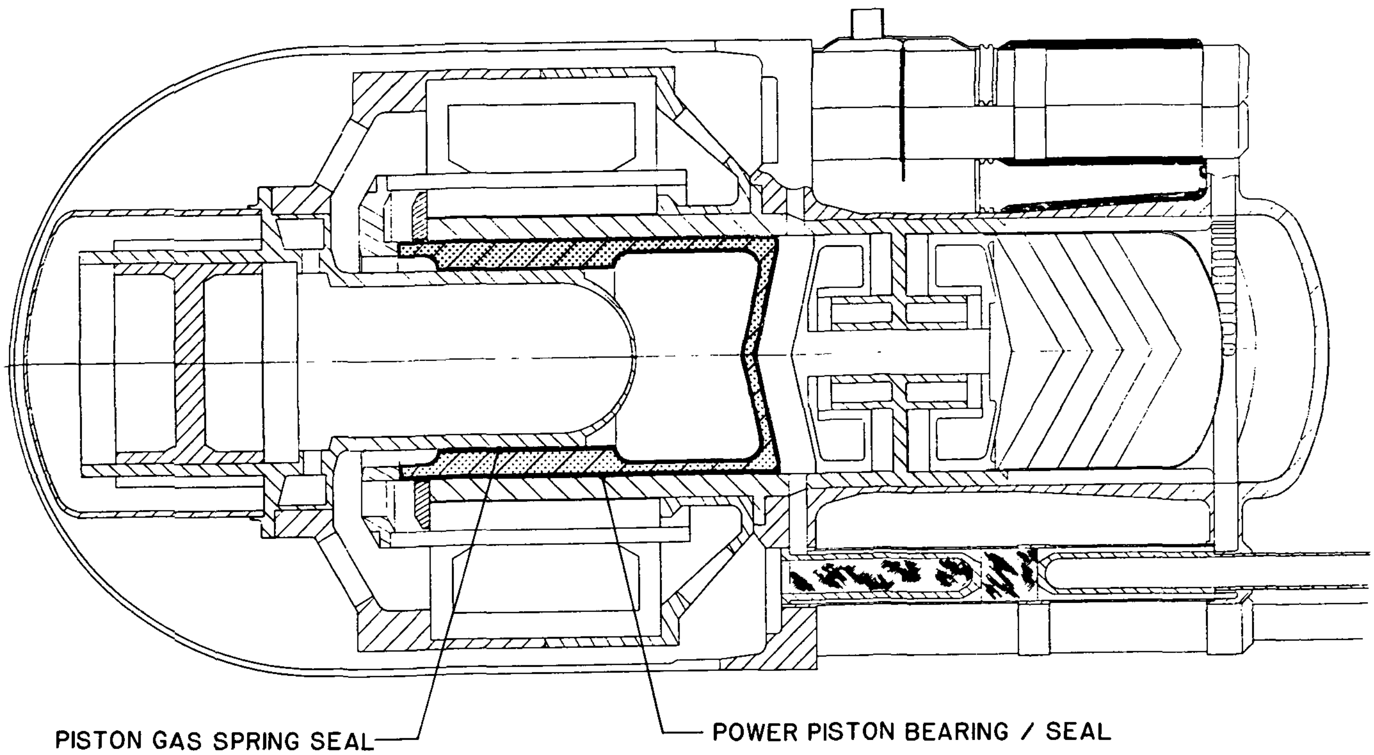


Figure 24: Power Piston

Due to its inherent dimensional stability, serious consideration was given to employing a ceramic for the power piston and cylinder. Of the materials considered, axially reinforced alumina or silicon nitride are the main candidates for the cylindrical section with a ceramic end closure forming the piston face. This cap would be attached via a ceramic-to-ceramic braze. Attachment of the alternator plunger structure to the piston represents the key fabrication issue since the conventional metallic materials employed have radically higher thermal expansion than the ceramic. One scheme would employ a compliant joint between the ceramic and metallic structure with some form of centering device, such as a spring, to maintain concentricity. The other, as shown in Figure 25, would employ a reinforced ceramic sleeve brazed to the cylinder which contains slots, akin to those employed on turbine blade roots, to support the magnet plunger. A mechanically attached ceramic end cap supports the unit and transmits alternator loads to the piston.

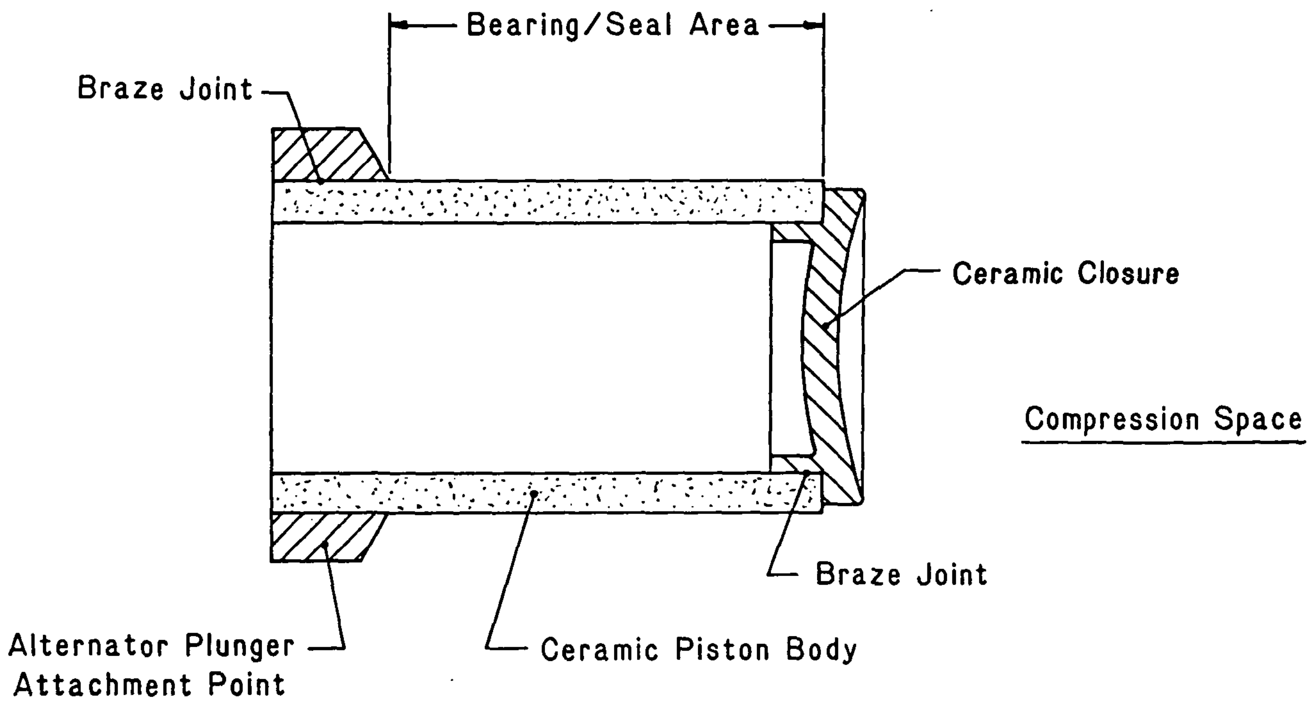


Figure 25: Possible Ceramic Piston Configuration

The displacer structure consists of two major components: the displacer shell, which makes up the forward portion of the displacer (hot end), and the combined gas spring/seal/bearing at the compression space end of the displacer. The unit is 160 mm in diameter and approximately 260 mm

long with a mass of about 5 kg. (See Figure 26.) The forward shell is an Inconel 718 structure that supports a number of internal radiation shields to minimize heating of the cooler beryllium portions of the displacer. This shell must withstand the engine pressure oscillations without structural instability. At present the primary failure mode is collapse caused by the portion of the cycle when the gas in the expansion space (outside the displacer shell) is at a higher pressure than the gas within it. The radiation shield support rings are employed to stiffen the structure under this condition.

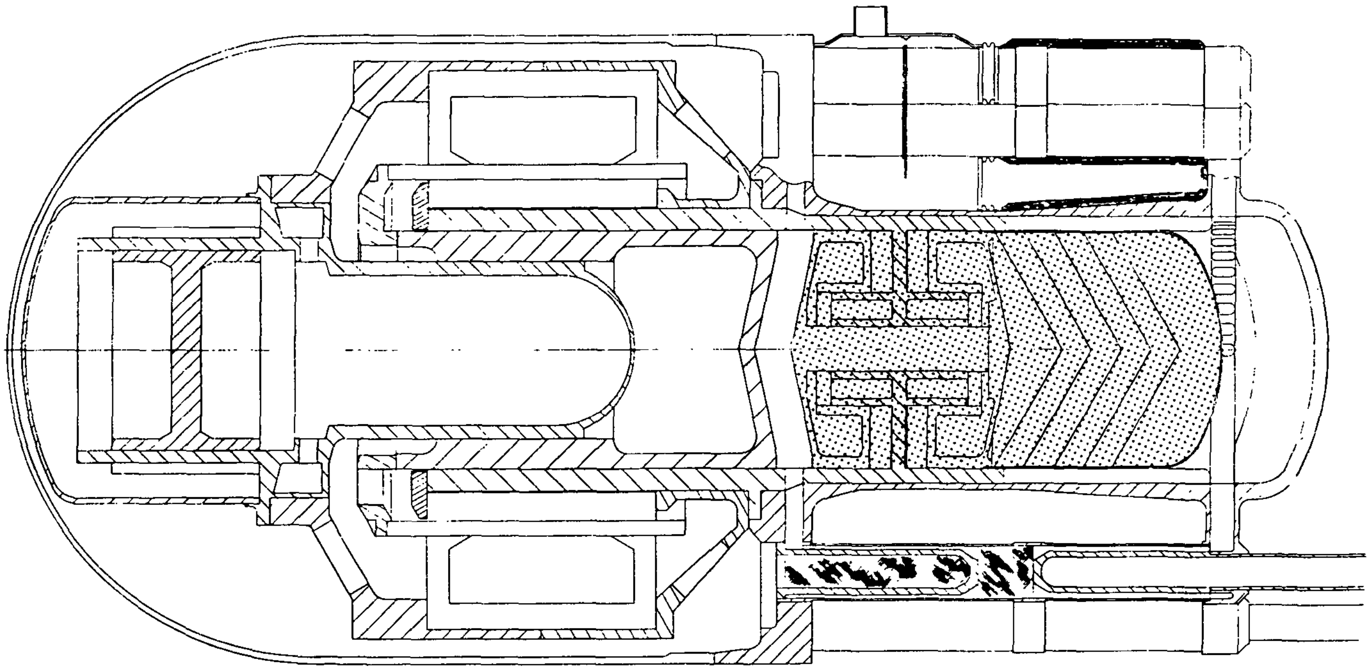


Figure 26: Displacer Structure

The rear portion of the displacer is an all beryllium structure that encloses two equal-strength gas springs that are required to properly tune the displacer dynamics. (This configuration is only one of a number of potential arrangements.) A total of five seals are required (See Figure 27.). The gas spring seals and bearing have a radial gap of 12.5 microns, while the cylinder seals separating the expansion and compression space employ a radial gap of 18.8 microns. This portion of the displacer represents the most complex machined component within the entire engine because of the number of concentric surfaces and the small gap dimensions. Turbines for spinning the displacer are attached to the displacer rear face. All beryllium seal and bearing surfaces employ hard surface coatings to provide protection for these critical surfaces during the initial spin up of the displacer assembly when some contact between the components can be expected.

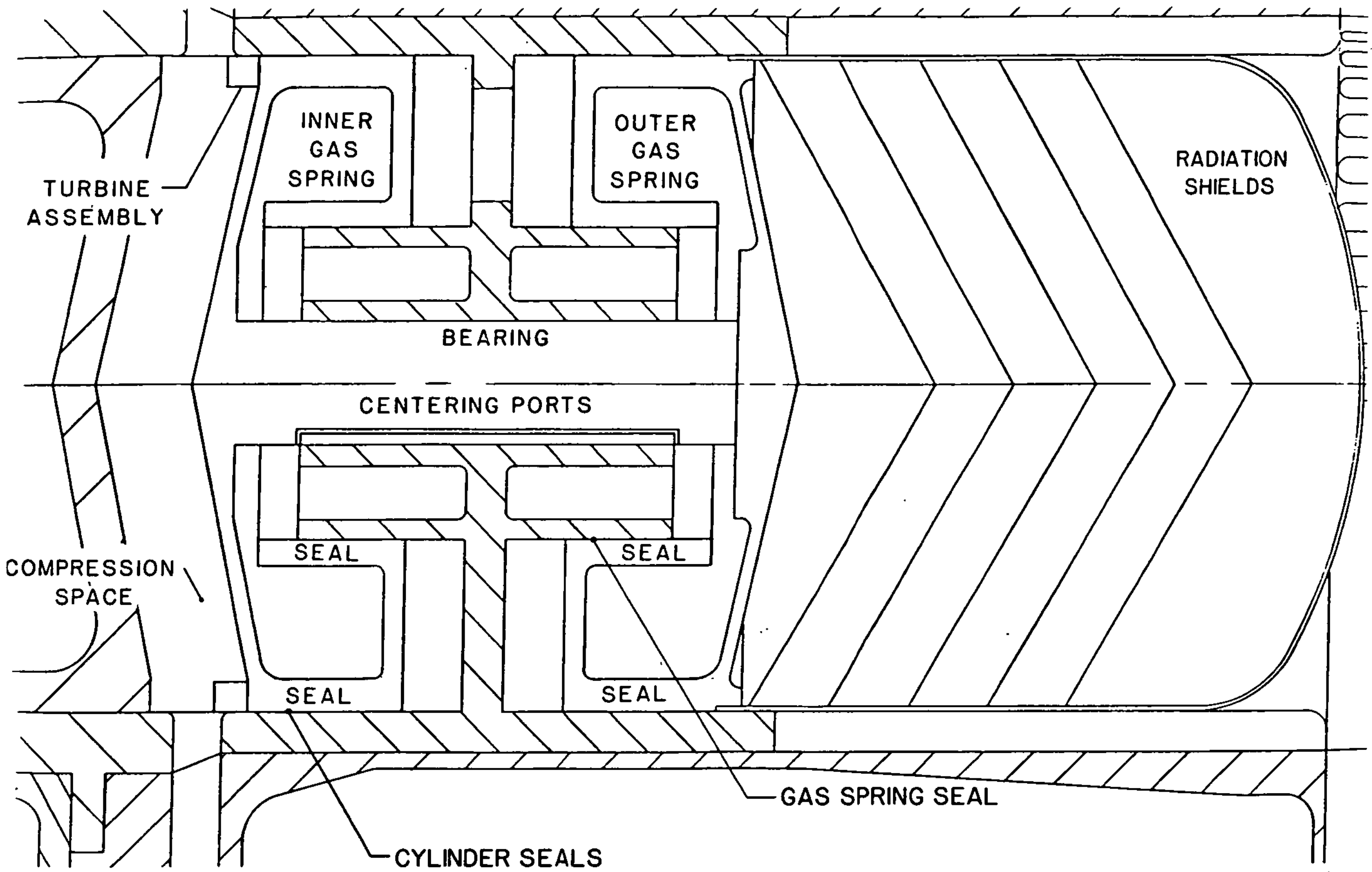


Figure 27: Displacer Mechanical Configuration

ALTERNATOR

The linear alternator is the key component that allows the inherent linear motion of the FPSE to be converted into electric power. Recent developments in linear alternator technology, particularly the advancements in permanent magnet materials and analysis techniques, allow the alternator to be quite compact and operate at conversion efficiencies in excess of 90 percent.

The current power module employs a conventional moving magnet linear alternator (See Figure 28.), which produces a net 27 kW of electric power at an overall efficiency of 93 percent.

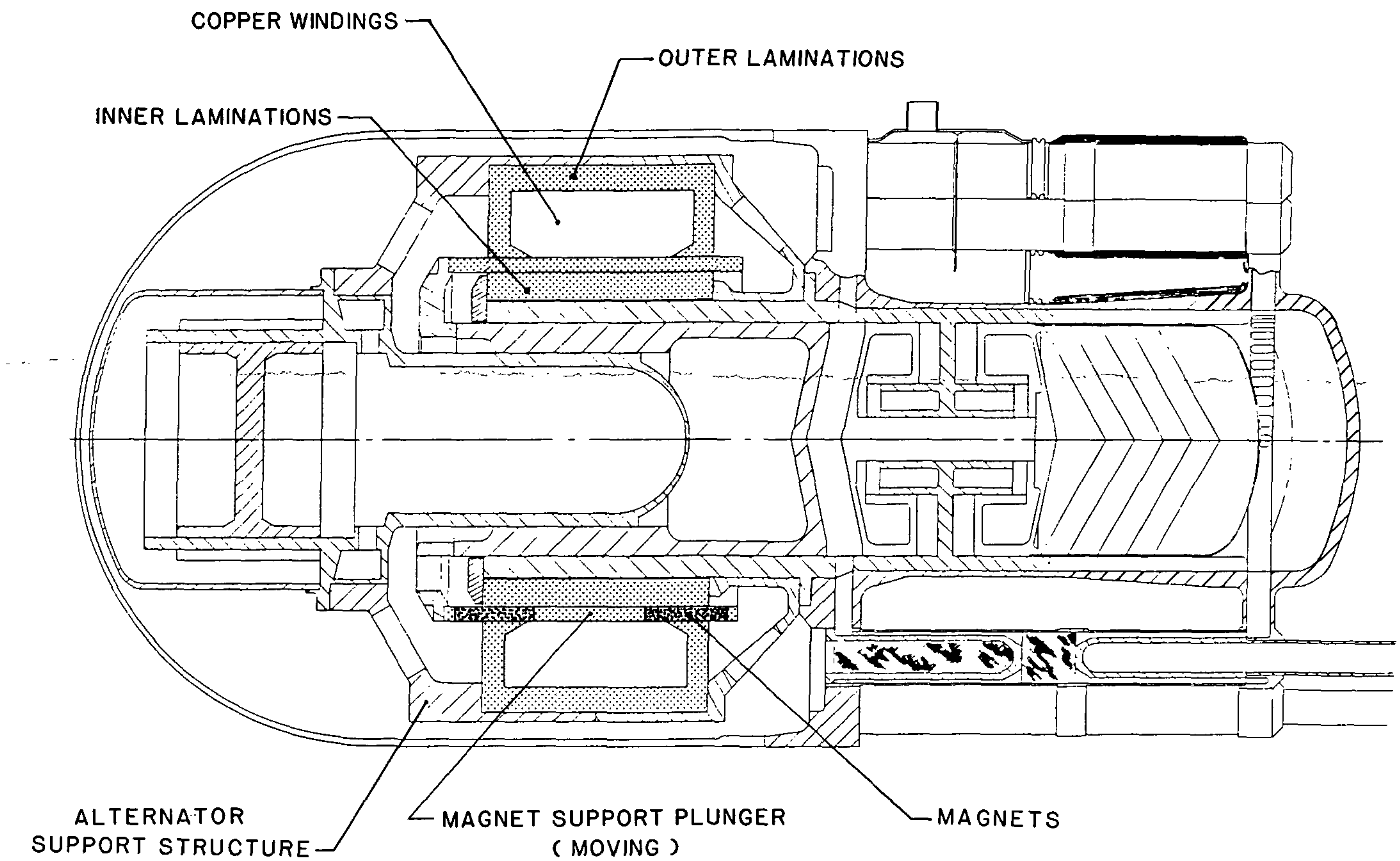


Figure 28: Power Module Linear Alternator

Key issues considered in the alternator design process are called out in Table 12. The moving magnet plunger is a composite structure supporting approximately 4 kg of samarium-cobalt material which is attached to the power piston via a titanium ring. Both the inner and outer stationary lamination sets are fabricated from a high cobalt-content alloy because of its excellent magnetic characteristics and are actually individual stacks of lamination rather than a continuous "doughnut-shaped" ring. This configuration makes more effective use of the lamination material while minimizing its mass. An inorganic coating is used for insulation on the copper coil which employs square wire to improve overall packing efficiency.

Table 12: Alternator Design Point

	Range Considered	Chosen Design Point
Frequency	90 to 110 Hz	90 Hz
Delivered Voltage	> 100 V	250 V
Specific Power	As high as possible	0.55 kW / kg
Delivered Power	> 27 kW	27 kW
Efficiency	As high as possible	93%
Piston Amplitude	10 to 20 mm	15 mm
Operating Temperature	As high as possible	525 K

Since the unit must operate at temperatures on the order of 525 K, an active cooling system must be employed to remove the waste heat generated by the alternator; without a cooling system, the magnet temperatures would rise above acceptable levels. With present permanent magnet materials, the 525 K operating temperature represents an upper limit if a conventional configuration of the alternator is used. In an effort to bypass this problem a more advanced alternator configuration was investigated during this program. This design, while somewhat more complex in configuration (See Figure 29.), minimizes the "loading" on the magnet material at high temperatures with the added advantage of requiring less external tuning capacitors.

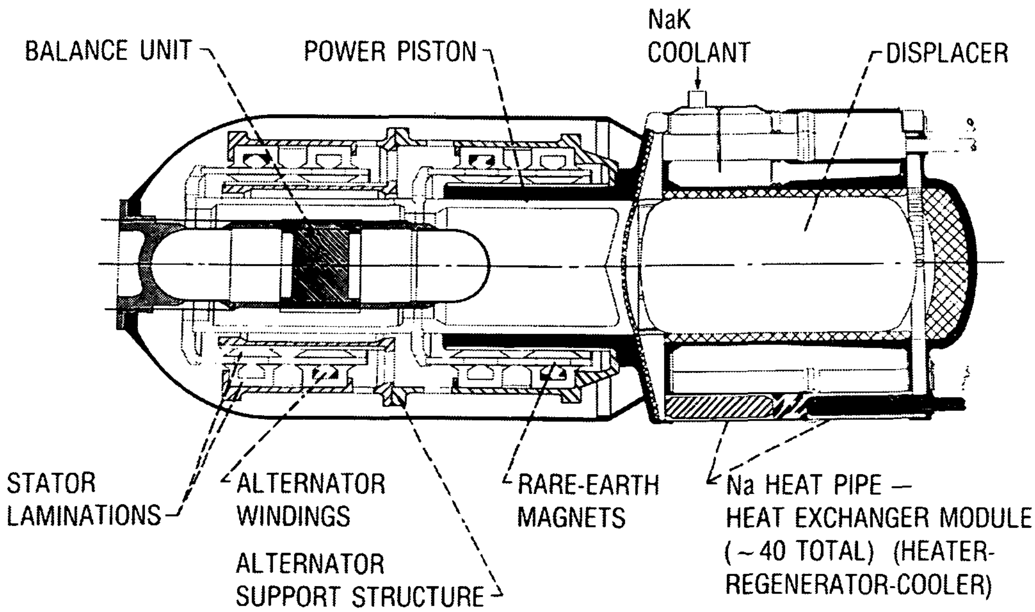


Figure 29: Advanced Alternator Configuration

In conclusion, the linear alternator represents a key element in the FPSE application to space power modules. While considerable effort has been expended in developing these devices, there are remaining technical and development issues (See Table 13.) that must be resolved before their full potential can be utilized.

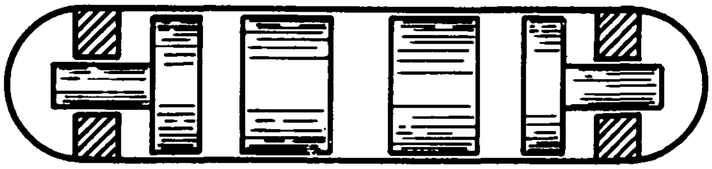
Primary among these is a better understanding of magnet life characteristics at power module temperatures coupled with electrical "loads" encountered in a high frequency linear alternator. Without these data, major decisions concerning auxiliary cooling requirements and power piston design cannot be adequately addressed. Interaction of surrounding structural members with the linear alternator electrical characteristics is a relatively straight forward process with current analytical and diagnostic tools; however, in the process of understanding these effects the possibility exists that major alternator geometry or material changes could occur, impacting power module design. More advanced alternator design work would also benefit from the efforts described above since a better understanding of the unique linear alternator loss interactions would be available.

Table 13: Alternator Issues Requiring Further Investigation

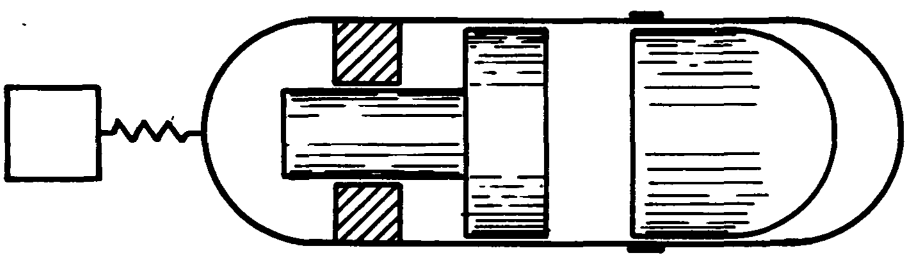
- Further Reduction or Elimination of Tuning Capacitors
- Improved Operating Temperature, Especially for Magnet Performance
- Improved Specific Power
- Reduced Complexity and Mass of Support Structures, Both Stationary and Moving, to Reduce Losses in These Support Structures
- Addressing Saturation Detuning Issue

BALANCE UNIT

Since the current unit is not of the opposed piston configuration (See Figure 30.), such as the SPDE, a net force is transmitted to the engine mounting structure due to the unbalanced loads induced by the moving piston and displacer.



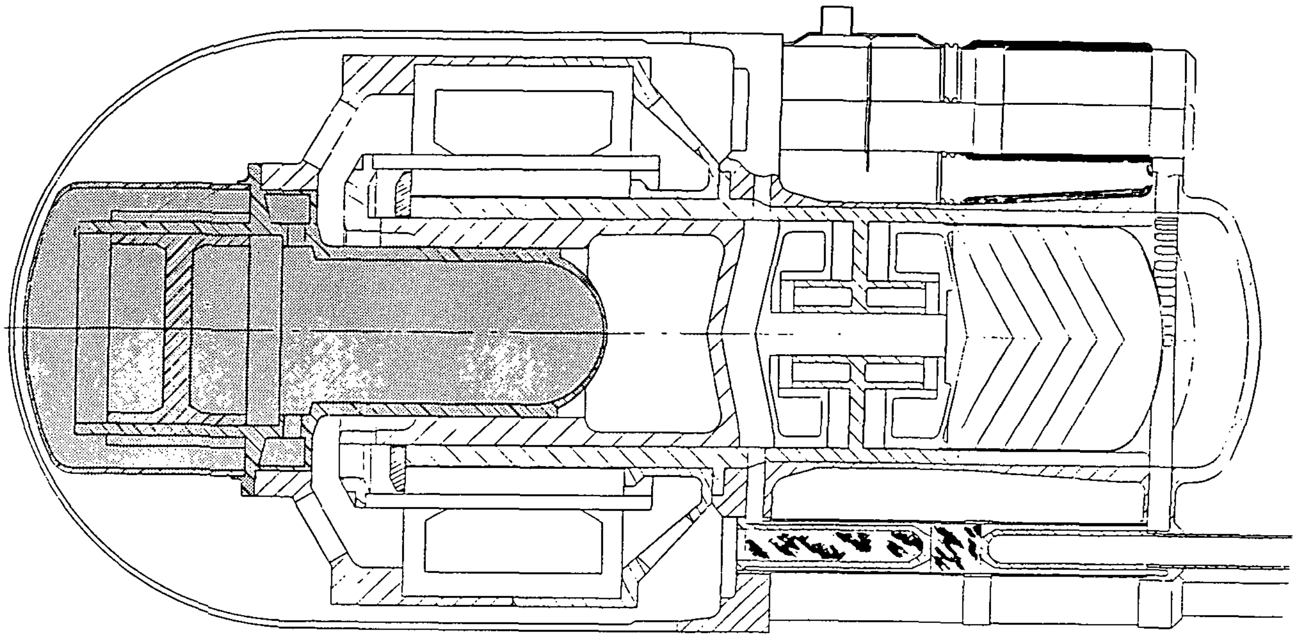
OPPOSED ENGINES



DYNAMIC BALANCE UNIT

Figure 30: Dynamic Balance Concepts

In conventional single-cylinder FPSEs, the casing is generally soft mounted via springs to its support structure. In the case of a spacecraft mounting, this does not provide enough isolation. To overcome this problem, the reference engine employs a third moving mass as a dynamic balance unit. This unit is located within the rear portion of the power module and is configured to act as a "plug" to form the power piston gas spring. The mass itself (See Figure 31.) oscillates within its own cylinder with gas springs on each of its faces. The mean pressure within the balance unit is the same as that of the engine so that the structure needs only to withstand the oscillating pressure forces of the gas springs. If the gas spring pressure were fixed at this value, it is possible that the balance unit could be out of tune with the engine during start up or other transient behavior; this would allow the transmitted forces to increase.



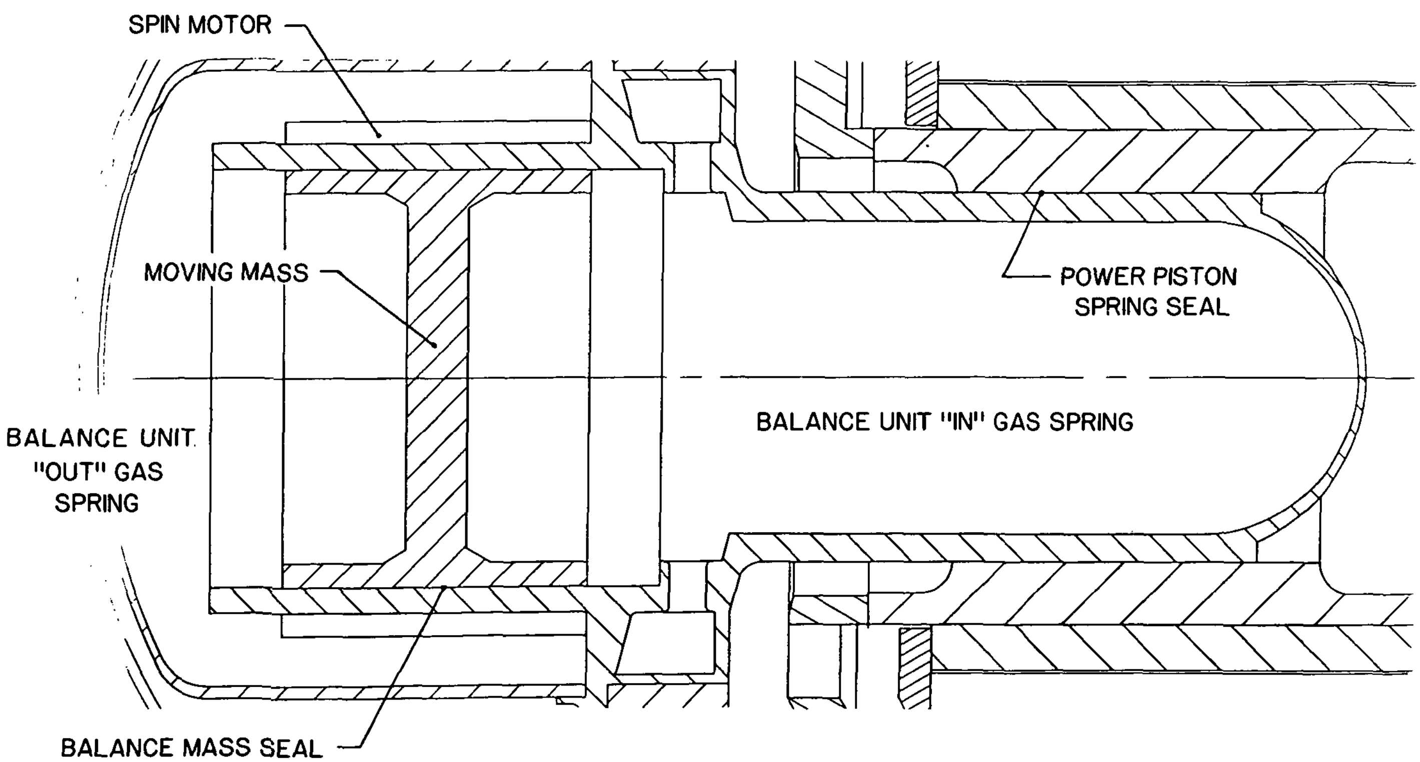


Figure 31: Dynamic Balance Unit

The present design has been evaluated with a fixed gas spring pressure as well as with a fully integrated control system which adjusts gas spring pressure to minimize measured acceleration of the casing, as shown in Figure 32. For the noncontrolled case, there is an increase during start up in net casing motion of about 35 percent compared with the controlled case. The duration of the overload is dependent on the specific engine start transient employed. Once the engine is at full power the nonconrolled configuration provides good balance performance assuming the engine remains within about ± 10 percent of its design pressure.

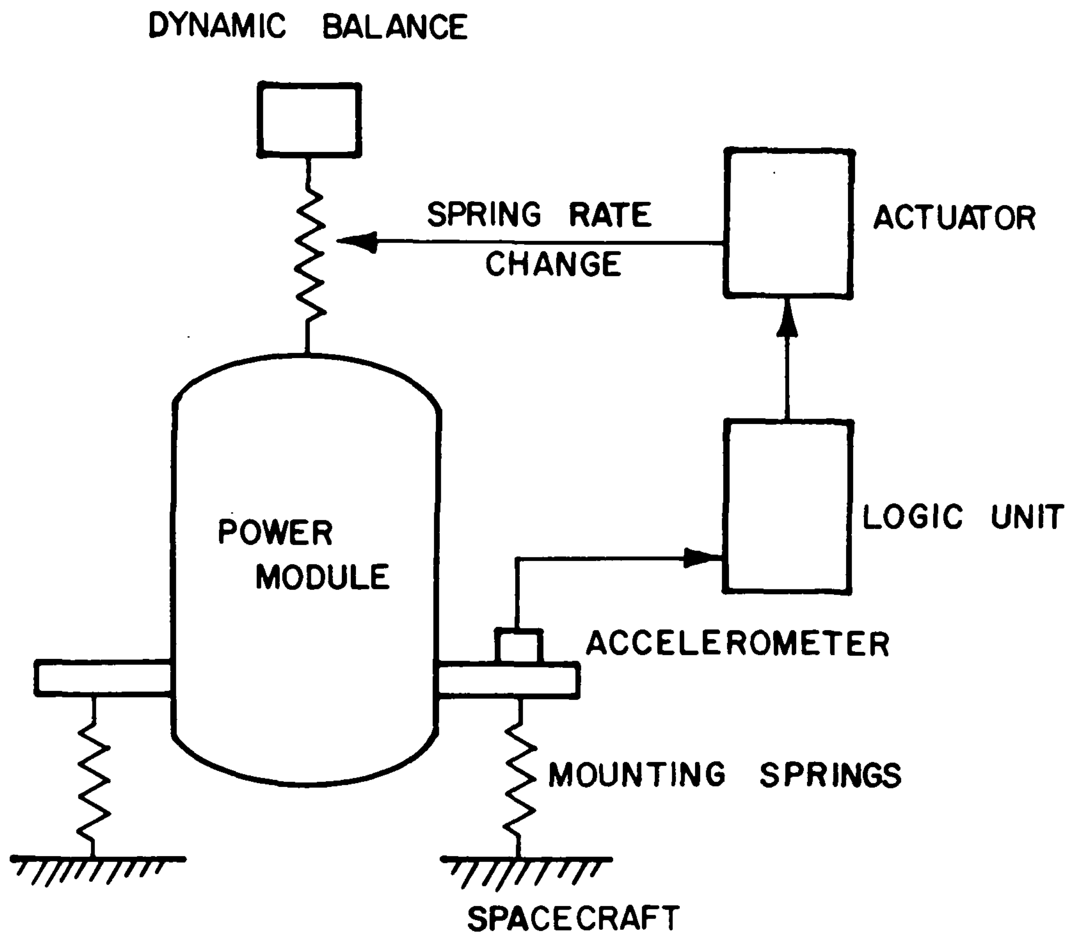


Figure 32: Active Dynamic Balance Unit Schematic

For the actively controlled balance system, the increase or decrease in gas spring pressure is accomplished by picking the pressure off the high or low side of the normal engine pressure oscillation (± 20 bar) in the compression space. Simulations indicate that this is more than enough to control the balance unit gas spring pressure over the range of operating conditions investigated. The operating point and basic characteristics of the balance unit are shown in Table 14.

For the actively controlled balance system, net casing motion is 35 microns. The balance unit does have disadvantages, however, which must be weighed against the opposed piston configuration. For the reference system the use of the balance unit causes a loss of approximately 690 watts of piston pV power and a module mass increase of 15 kg (about 9 percent of total). In the current system, these effects have been minimized by placing the unit within the pressure vessel itself and by selecting a reasonable compromise between the mass of the moving component and the combined seal leakage losses and gas spring hysteresis. Reducing the moving mass requires larger amplitude which, in turn, causes more leakage and hysteresis losses; the reverse is true if the moving mass is increased.

To maintain noncontact operation between the balance unit moving mass and its cylinder, the moving mass is spun via an electric motor powered off the linear alternator. This essentially introduces a "third" piston into the single-cylinder system, somewhat reducing its advantages in comparison to the "four" pistons in an opposed piston design. Also, as mentioned above and shown in Table 14, there is a performance penalty which can reduce overall system efficiency and increase the quantity of heat that must be rejected from the alternator space by about 35 percent.

Table 14: Dynamic Balance Unit

Operating Point

Balance Unit Moving Mass	12 kg
Mass Diameter	0.136 m
Balance Unit Mass Amplitude	0.0183 m
Net Casing Motion	3.5×10^{-5} m
Gas Spring Hysteresis	435 W (Total)
•Gas Spring	$\Delta/V \simeq 0.077$
•Pressure Amplitude	~ 19.4 bar
Moving Mass Leakage Losses	255 W
•Radial Gap	1.25×10^{-5} m

GROUND-BASED EXPERIMENTAL ENGINE

A ground-based test engine based on the existing power module design was briefly investigated as a portion of the current project. The basic goals for such an engine are listed in Table 15. Since the unit is a test engine and as such requires replacement or modification of various internal components, it would need a separation flange in the area noted in Figure 33. Also, since the balance unit represents a critical component, it has been moved to the outside of the pressure vessel. This will require that its pressure-containing walls be increased in thickness and a separation flange added; however, all internal components would remain the same (moving mass and spin bearing). Other changes in comparison to the reference engine are listed in Table 16.

Table 15: Ground-Based Experimental Engine

Goals

- Demonstrate All Key Elements of SSE Design
  with Possible Exception of Design Life at Design Temperatures
- Achieve Hot-End Design Point 1000 K to 1050 K
- Address Reliability, Ease of Component Modification,
  Ease of Assembly and Disassembly
- Provide Instrumentation Access as Possible
  - To Measure Critical Engine Parameters
  - To Determine Engine Loss Mechanisms
- Track Changes from Reference Design that Affect Engine Mass

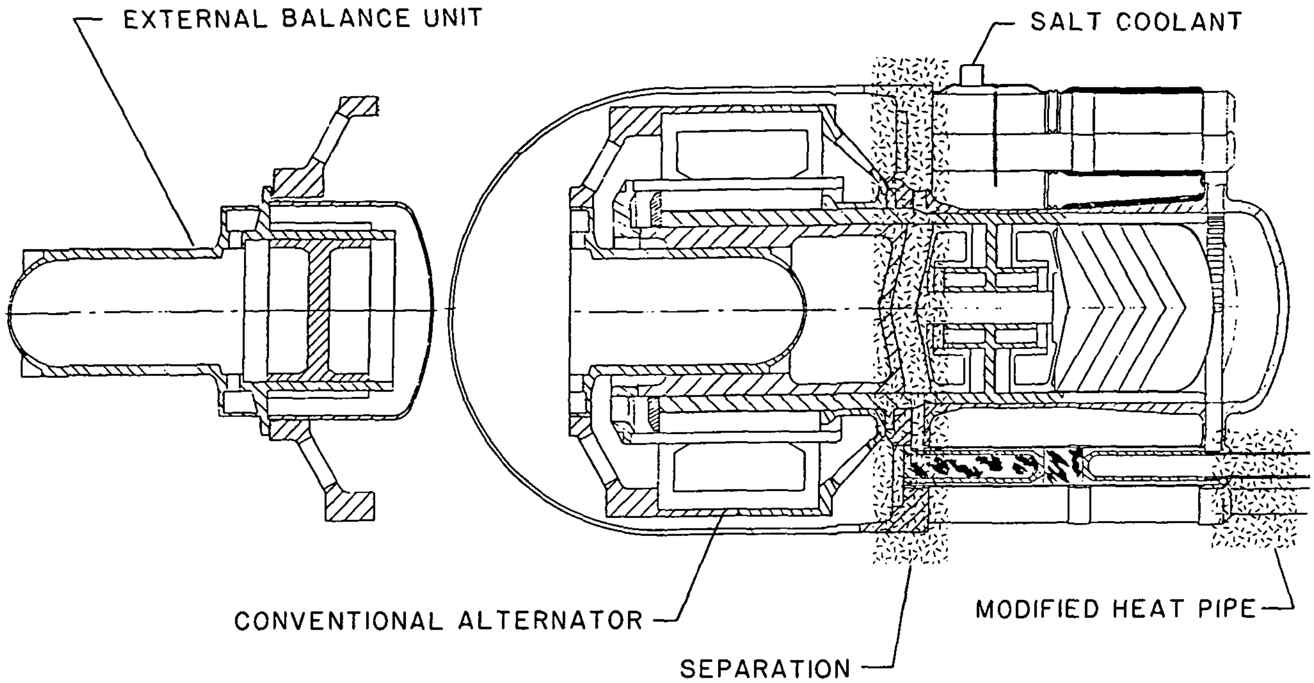


Figure 33: Ground-Based Test Engine

Table 16: Ground-Based Test Engine

Differences from Reference Engine

- Wrought Inconel Alloy Hot-End Section
- Limited Operating Life Capability at 1050 K
- Externally Mounted Balance Unit
- Salt or Oil Coolant Instead of NaK
- Replaceable Heat Exchanger Modules as Possible

CONCLUDING REMARKS

The Stirling Space Engine (SSE) design indicates that free-piston Stirling engines employing linear alternators represent an attractive power conversion unit for space power systems. This single-cylinder design was optimized for the lifetimes (seven years) and temperatures appropriate for space applications. It offers long life and high reliability with only three noncontacting moving parts and a hermetically sealed system. The high efficiency (29 percent) and low specific mass (5.7 kg/kW(e)) at a temperature ratio of 2.0 makes it a viable space power conversion unit at its design temperature of 1050 K. The design also represents a significant step toward the 1300 K refractory Stirling engine; this is a growth option for the SP-100 power system offering increased power output and lower system mass and radiator area.

A good understanding of the materials characterizations and fabrication issues will be vital to the fabrication of the SSE based on this reference design. The primary materials/fabrication issues are: 1) friction welding of Inconel 713; 2) achieving sodium compatibility and strength while maintaining low temperature differences across the wall for the hot-end heat exchanger; 3) maintaining clearances of the piston and displacer; 4) minimizing losses to the support structure of the alternator; and 5) achieving magnet life at the required cold-end temperature. The key component areas in the design are the heat pipe/heat exchanger modules, the hydrodynamic gas bearings, and the linear alternator.

Finally, the trade-offs between the single-cylinder and the opposed-piston designs should continue to be reviewed. The opposed-piston design does not require a separate balance unit and thus, the additional mass and losses associated with the balance unit could be eliminated. However, the opposed-piston design will probably require more complex heat pipes and possibly a more complex system to transport heat to the engine.

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15. Supplementary Notes

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16. Abstract

As part of the NASA CSTI High Capacity Power Program on Conversion Systems for Nuclear Applications, Sunpower, Inc. completed for NASA Lewis a reference design of a single-cylinder free-piston Stirling engine that is optimized for the lifetimes and temperatures appropriate for space applications. The NASA effort is part of the overall SP-100 program which is a combined DOD/DOE/NASA program to develop nuclear power for space. Stirling engines have been identified as a growth option for SP-100 offering increased power output and lower system mass and radiator area. Superalloy materials are used in the 1050 K hot end of the engine; the engine temperature ratio is 2.0. The engine design features simplified heat exchangers with heat input by sodium heat pipes, hydrodynamic gas bearings, a permanent magnet linear alternator, and a dynamic balance system. The design shows an efficiency (including the alternator) of 29% and a specific mass of 5.7 kg/kW. This design also represents a significant step toward the 1300 K refractory Stirling engine which is a growth option for SP-100.

17. Key Words (Suggested by Author(s))

Stirling engine; Heat engine;
Space power; Linear alternators;
Gas bearings; Heat pipes

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